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Authors

Danny S. Parker
John R. Sherwin
Bart Hibbs

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A Research Institute of the University of Central Florida
1679 Clearlake Road, Cocoa, FL 32922-5703 • Phone: 321-638-1000 • Fax: 321-638-1010
www.fsec.ucf.edu

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Danny S. Parker and John R. Sherwin
Florida Solar Energy Center

Bart Hibbs
AeroVironment, Inc.

Abstract

With sponsorship from the U.S. Department of Energy, a research project has designed, fabricated and tested improvements to an air conditioner outdoor unit fan system. The primary objective was to improve condenser fan performance while reducing motor power. We also examined potential changes to the condenser exhaust configuration to enhance air moving efficiency performance. A secondary objective was to provide sound reductions as lower noise AC equipment is important to consumers.

Within conducted tests, an improved high efficiency fan design and advanced exhaust diffuser section reduced fan motor power requirements by approximately 49 W (26%) while providing superior air flow. When mated with a brushless DC motor, the same configuration can reduce fan power use by nearly 100 Watts (50%). The overall increase to total system efficiency (EER and COP) is approximately 2-4% depending on configuration. The reduced fan unit power could be very desirable for utilities concerned with peak demand, since the change provides reliable load reductions on peak.

The changes in exhaust configuration are also important in that they allow for slower fan speeds to obtain equivalent flow. When coupled with a developed vortex shedding control strip and an asymmetrical fan design we showed reductions to fan sound levels of 1-2 dB according to ARI Standard 270-1995.

Introduction

Air-cooled condensers in residential air conditioning (AC) systems commonly employ finned-tube construction to transfer heat from the refrigerant to the outdoor air. As hot refrigerant passes through the condenser coil, heat in the compressed refrigerant is transferred through the tubes to the attached fins. An electrically powered fan draws large quantities of outside air across the finned heat transfer surfaces to remove heat from the refrigerant so that it will be condensed and partially sub-cooled prior to its reaching the expansion valve. A conventional AC condenser and fan is illustrated in Figure 1.

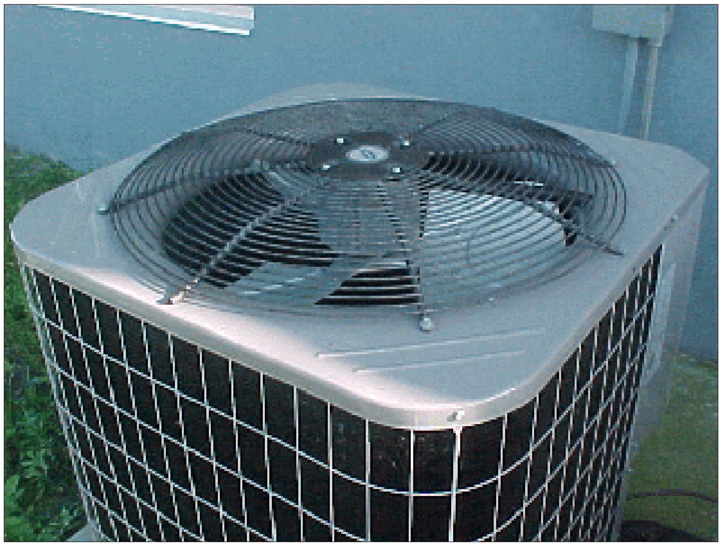

Figure 1. Typical 3-ton (10.6 kW) air conditioner condenser

The air conditioner condenser fan is one energy using component of a residential air conditioning system. The largest energy use of the air conditioner is the compressor. Intensive research effort has examined improvements to its performance. The other components are the indoor and outdoor fans. However, much less effort has examined potential improvements to the system fans. These include both the indoor unit fan and that of the outdoor condenser unit.

Residential air conditioners are a major energy using appliance in U.S. households. Moreover, the saturation of households using this equipment has dramatically changed over the last two decades. In 1997, for instance, 73% of U.S. households had air conditioning as opposed to 56% in 1978 (DOE/EIA, 1999). The efficiency of residential air conditioners have large impacts on utility summer peak demand. Thus, improved efficiency of air conditioning systems is both desirable for consumers as well as utilities. One advantage of the proposed research is that it will have small, but identifiable impacts on system performance under peak demand conditions.

For instance, if the condenser fan motor can be reduced in size from 1/4 hp to 1/8 hp, the approximate drop in peak AC demand could amount to 100-150 Watts, depending on motor efficiency. This translates directly to reduced AC unit utility peak demand.

Background

The outdoor condenser fan draws a large quantity of air (2,000 - 4,000 cfm or 944-1,888 L/s) at low static pressure – typically 0.05 to 0.2 inches of water column (IWC) (12-50 Pa) through the condenser coil surfaces. A typical 3-ton (10.6 kW) air conditioner with a seasonal energy efficiency ratio (SEER) of 10 Btu/Wh (SCOP = 1.93 W/Wh) moves about 2,400 cfm (1,133 L/s) of air using about 260 Watts of motor power. The typical outdoor fan and motor combination is an axial metal propeller type fan inserted in a short diffuser with a fan efficiency of about 20% - 25%; and a permanent split capacitor (PSC) motor with a motor efficiency of about 55% - 60%.[1] Typically, a 1/4 hp (0.19 kW) motor would be used for a three ton air conditioner. The resulting air "pumping efficiency" is roughly 11 - 15%. Lower condenser fan electrical use is now available in higher efficiency AC units through the use of brushless direct current (BDC) permanent magnet motors and larger propellers.[2] These changes improve the combined fan and motor air moving efficiency, but only about 20-25%.

A literature survey revealed a number of studies evaluating air conditioner condenser performance, but few examining specific improvements to the outdoor fan. One investigation did identify larger condenser fans as potentially improving the air moving efficiency by a few percent (Proctor et al., 1994). The same study also identified the need for more efficient fan blade designs although it did not undertake that work.

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[1] Motor efficiency is the input energy which the motor converts to useful shaft torque. Fan efficiency is the percentage of shaft torque which the fan converts to air movement.

[2] Commercially the brushless DC motors are often called electronically commutated motors or ECMs. Although the more efficient BDC motors gaining in the AC market, these are more expensive. For instance the replacement retail price of a standard 1/4 horsepower (0.19 kW) PSC condenser fan motor might cost $80 whereas a similar more efficient BDC motor might cost $310.

Currently, major air conditioner manufacturers are working to maximize the performance of conventional air conditioners to increase cooling system efficiency in a cost effective manner. Within this effort every watt of cost-effective power reduction is potentially important. In this project we explored how improvements to the outdoor unit propeller design as well as reduction of the external static pressure resistance of the fan coil unit could have large overall impacts on potential efficiency.

Conventional fan blades used in most AC condensers are stamped curved metal blades that are inexpensive to manufacture, but may not be optimized in terms of providing maximum air flow at minimum input motor power. See Figure 2.

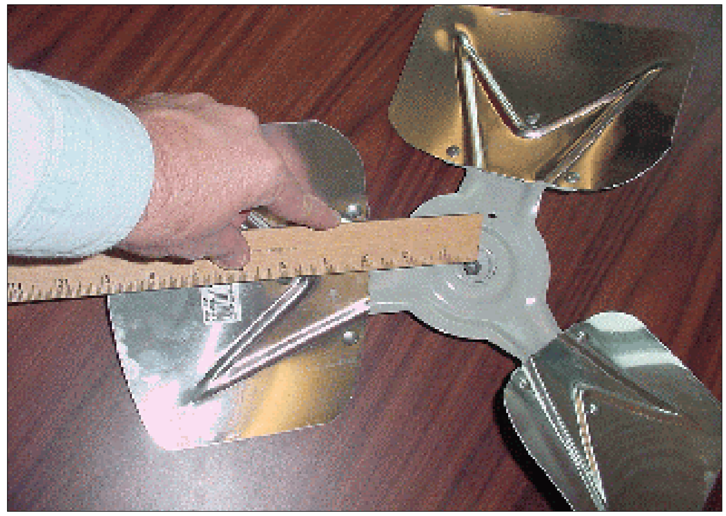

Figure 2. Stamped metal condenser fan blades for a 3-ton condenser

For instance, a typical 3-ton (10.6 kW) condenser fan from one U.S. manufacturer draws about 260 Watts for a system that draws 3,750 Watts overall at the ARI 95/80/67 test condition (ARI, 2003). Thus, potentially cutting the outdoor fan energy use by 30% - 50% has the potential to improve air conditioner energy efficiency by 2 to 3%.

Our research attempted to design fan blade shapes with true airfoils that would fit in conventional AC condensers (e.g. 19 inches or 48 cm wide for a three-ton condenser (10.6 kW) and 26" (66 cm) wide for a higher efficiency model). At the same time, they were to provide the best possible air flow at a design rotational speed of 850 rpm with the lowest possible shaft input wattage.

Fan Performance

Air conditioner fans, like all fans and propellers, are governed by fan laws. Key implications of these laws are summarized in the bullet items below and graphically illustrated in Figure 3.

- The power required to move air increases at the cube of the quantity of the mass flow. This exacts a real limit to the air flow that can be realistically increased to improve coil heat transfer.

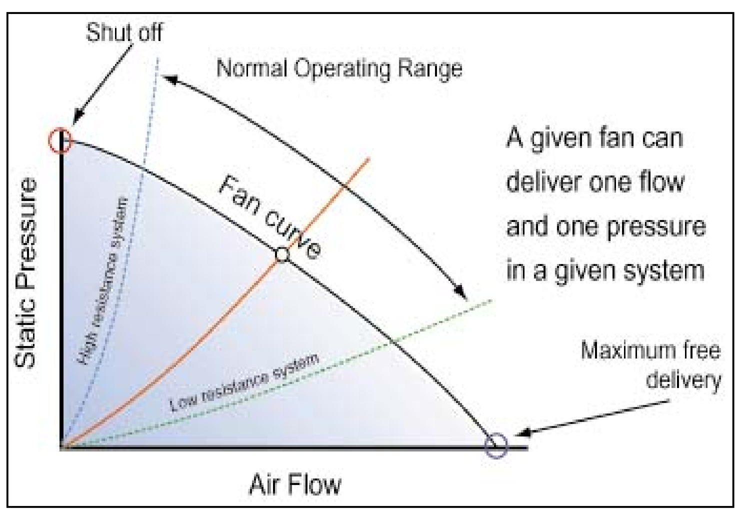

Figure 3. Illustration of fan law impacts on air flow in AC condenser

- The system static pressure (negative pressure on the underside of the fan and over-pressure above the fan) increases at the square of the air flow increase.

- As static pressure decreases, the air flow increases along the fan curve for a given fan.

- Fan curves can have peak efficiency and stall regions with implications for air moving performance and sound.

- With a given fan and system pressure, the air mass flow rate increases linearly with fan RPM.

- PSC motors offer limited RPM selection: 825-850 rpm for 8 pole motors, 1,075-1,100 rpm for 6 pole motors and 1,450-1,500 rpm for 4-pole motors. Slower motors are preferred due to sound implications.

- Brush-less direct current (BDC) permanent magnet motors offer variable speed control to select the RPM range over the range to which it is programmed. Thus, these motors allow flexible choice of condenser fan air flow without sacrificing efficiency.

One challenge within the research was to adequately measure the external static pressure of the fan as operating within the condenser. As shown in Figure 3, establishing this value was critical to the fan and the achievement of good performance. Within our testing, this was measured under the fan, taking a traverse of the condenser cavity using a precision digital manometer. Measurement of the pressure above the fan and under the grill proved more difficult, but sufficient data was obtained to facilitate a fan design. The total external static pressure of the original fan producing ~2,200 cfm (1,038 L/s) was about 30 Pa (0.12 IWC).

Impact of Air Flow on Outdoor Unit Condenser

As expected, greater air flows across the condenser coil heat rejection surfaces lead to greater cooling capacity and slightly lower compressor power. A simulation was run implementing the DOE/Oak Ridge Heat Pump model to examine the relationship between motor power and airflow for a conventional 3-ton cooling system (Fischer and Rice, 1985). This simulation model has been well validated (Levins et al., 1997 and Rosenquist, 1997). It allows detailed examination of how system cooling performance is enhanced by increased air flow across the condensing surfaces and how this trades off against increase to condenser fan motor power. Figure 4 shows the relationship evaluated for a conventional AC system.

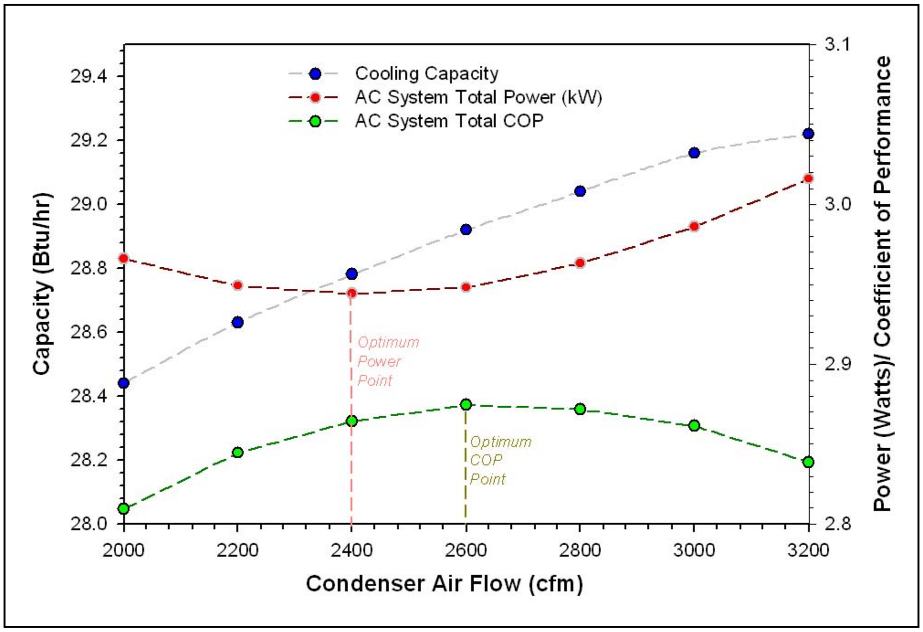

Figure 4. Evaluation of Optimum System Outdoor Air Flow for a Hypothetical 3-ton air conditioner

Note that the slope of the increased cooling system capacity increases gradually with greater air flow, while the required fan power increases rapidly with flow. As the required shaft power will increase between the square and the cube of the air flow, this exacts a real limit on the air flow to be selected for a given condenser design. Thus, the overall system electric power (including compressor, and indoor and outdoor fans has an optimum outdoor unit air flow where electric power is minimized and another point where the system cooling coefficient of performance (COP) is maximized. These two points are close and between 2,400-2,600 cfm (1,133-1,227 L/s) for the unit modeled. Not surprisingly, real AC units of this size often have rated air flow of around 2,400 cfm (1,133 L/s).

It should be noted, however, that the optimum condenser coil face velocity and condenser air flow is affected by the coil face area, coil depth and fin spacing (Green and Roberts, 1996). Generally, the highest COPs are achieved for coils with a large face area, operating at a high evaporator temperature and a low air velocity (2 m/s). However, as we had no control over the heat exchange surfaces in the AC condenser in order to lower external static pressure, the challenge for our research was to design a fan and exhaust manifold for our test air conditioner which would improve the overall air moving efficiency and maintain or slightly increase flow.

Sound Control

Although improving condenser fan energy efficiency was the fundamental goal of our research, a secondary objective was to reduce sound levels due to the importance consumers place on a quiet air conditioner. This is particularly important with close lot lines where AC condensers can be near to sleeping quarters in neighboring houses.

Sound is measured in decibels (dB) above the background noise level. As the scale is logarithmic, small changes can mean large changes in sound level. For instance, 1 dB is generally accepted as the minimum sound level difference that people can discern. A change of 3 dB (20% change in sound pressure) is noticeable and a 5 dB difference is clearly noticeable. ARI Standard 270-1995 governs the way in which sound levels are measured for outdoor AC units.

The importance of the sound issue is clearly illustrated by a survey done of 550 individuals in Canada (Bradley, 1993) which found that complaints from air conditioner noise dramatically increased when the sound level was 5 dB or more above background levels. Also, the same survey found that homeowners expressed a willingness to pay 12% more for a very quiet air conditioner.

Although the topic of sound and vibration control within air conditioning is quite complex, we describe here some of the fundamental influences (Schaffer, 1991):

- Fan rotation speed is a major factor in sound propagation

  $$dB_1 = dB_2 + 50\log_{10} (rpm_1/rpm_2)$$

  Thus, a fan moving at a 20% slower speed should exhibit a 5 dB drop in sound level

- Other factors:
  - Vortex shedding: turbulent eddies in the wake of the fan blade tips
  - Turbulence due to the obstructions in the intake or exhaust wake
  - Fan motor vibration
  - Harmonic resonance associated with the number of blades
  - Fan interaction with compressor noise

Within our research, a key emphasis to reduce sound levels was to operate fans more slowly with efforts made to suppress fan tip vortex shedding and harmonic resonance associated with fan blades.

Baseline Air Conditioning Unit and Test Facility

For our testing, we used a standard 3-ton (10.6 kW) air conditioning system produced by a major U.S. manufacturer. The system uses R-22 refrigerant, although our evaluation was done with condenser fan only operation. The system has a rated SEER of 12 Btu/Wh (SCOP = 3.51 W/Wh) when mated with a compatible evaporator and air handler. The 19" (48 cm) fan in the original outdoor unit consists of four metal paddle blades, powered by a six-pole 1/8th hp (0.09 kW) motor with a rated flow of 2,400 cfm (1,133 L/s) for the condenser. As measured in the baseline condition, the fan motor drew 190-197 Watts at 208 Volts and produced 2,180-2,200 cfm (1,029-

1,038 L/s) turning a 1,010 rpm. At the 95/80/67 ARI test condition the fan power for the entire AC system represents about 6% of total system power.[3]

For the research, we needed to accurately measure power, condenser air flow, fan motor power and rpm as well as environmental conditions. Secondly, we desired to measure sound levels. For diagnostic purposes, we also used flow visualization tools (smoke pencils and flow wands) to aid our understanding of the air flow dynamics.

An indoor test facility was constructed. A precision power transducer provided motor power measurements (±1 Watt resolution) and air temperature was obtained by a condenser inlet thermo couple. As the facility had 208 single-phase power, this electrical source was used for the measurements involved. A laser tachometer was used to measure fan rpm and a precision portable dB meter was used to measure nearby sound levels.[4] A digital manometer was used to measure static pressure within the condenser underneath the exhaust fan. As condenser air flow was a critical measurement, we constructed flow measurement chamber in conformance with ASHRAE Standard 51-1985. The constructed outlet duct chamber with flow straighteners and settling screens was then calibrated at another facility with NIST traceable air flow equipment. The final chamber was estimated to yield an absolute air flow measurement accuracy of approximately ± 5% (125 cfm or 59 L/s). The relative air flow measurement accuracy was much better. We found the equipment could reliably measure changes in air flow as small as 20 cfm (9 L/s) out of a 2,500 cfm (1,180 L/s) air flow.

Development of New Fans

In designing fans, our objective was to create the most efficient designs while operating at low rotational speed to reduce fan noise. We also looked to create robust characteristics which would provide good performance over a range of static pressure. This is important because static pressure can increase due to condenser fouling or frost buildup during winter operation. Generally, the true airfoils we used have flatter fan curves than those for curved metal bladed fans.

Over a period of two years, a total of five different fans were designed and built (designated A-E) with a series of sub-variations on each. The differing fan configurations were targeted for differing rpm ranges, static pressure rise and sound characteristics. The Original Equipment Manufacturer (OEM) design was a stamped 3-bladed metal fan. Fans A and D consisted of three equally spaced blades, with tapered and twisted air foils. Fan D was designed for a higher

[3] Improvements to a higher-efficiency condenser using R-410A with a nominal system SEER of 14 Btu/Wh are also being evaluated. This condenser is physically larger with also a larger fan (27.6" or 70 cm) and a 1/4 hp (0.19 kW) PSC motor. Within this configuration, fan power (240 Watts) represents 8% of overall system power at the ARI 95/80/67 condition.

[4] We did not have the equipment and acoustic room to measure sound levels in compliance with *ARI Standard 270-1995*, but did measure at the prescribed height and distance with a tripod mounted dB meter. This gave a relative indication of design changes on sound. However, later the ARI sound impact was measured in an sound laboratory in compliance with prescribed procedures.

pressure rise. Fan A5 was an asymmetrical 5-bladed design. Fan E had forward curved blades, intended to assist with sound reduction. Each of the designs were evaluated by computer simulation and then produced as three dimensional coordinate files that could be used to describe the complex shapes. Rapid prototyping was used to physically produce the fan blades. Each fan was then hand-mounted onto a produced hub and speed balanced before evaluation on the test stand.

The fan design with an asymmetrical alignment has unequally spaced blades. This technology was explored to potentially lower noise levels. This technology has been previously developed for helicopter rotors (Kernstock, 1999), but not previously utilized for AC condenser fans. The sound of air rushing through an evenly spaced fan rotor creates a resonance frequency with the compressor's hum, leading to a loud drone. But if the blades are not equally spaced, this resonance is reduced producing lower ambient sound levels. With our invention, we took advantage of the asymmetrical characteristics using a five-bladed fan design where the fan blades are centered unevenly around the rotating motor hub (Figure 5).

Figure 5. 19" Fan A5 with asymmetrical blade spacing.

We tested each fan design with 1/8 hp (0.09 kW) PSC motors either with six poles rotating at approximately 1075 rpm or eight poles rotating at 850 rpm. Over one hundred tests were conducted over an 18 month period.

Diffuser Design

Diffusers are an expanding duct which provides recovery of air static pressure by reduction of the flow velocity as the flowing air mass expands. Practically, the condenser fan air velocity is lowered prior to exhaust, thereby increasing the overall mass flow rate from the system or conversely reducing the load on the motor to produce a given flow. The exhaust configuration of a standard unitary air conditioner consists of a short 4", 10 degree divergent diffuser covered by a slotted grate or wire grill with the fan is nestled in the bottom of the diffuser section.

Examining interactions between high efficiency propeller designs and external static pressure, we determined that an optimized diffuser section would allow large improvements in air moving efficiency. Diffuser theory would suggest that large improvements in fan efficiency are possible by lengthening the diffuser stage (Blevins, 1984). Theoretically, an 18" (46 cm) diffuser should provide about 25% added pressure recovery over that from a short 4" (10 cm) diffuser (Japiske and Baines, 1993). While a longer length would provide still greater pressure recovery, we judged an 18" (46 cm) height to be the maximum practical for consumer acceptance.

Thus, we constructed a larger 18" (46 cm) tall 7^{o} divergent diffuser with the motor and fan located in the bottom of the assembly. Figure 6 shows a schematic of the overall assembly as produced with the elongated diffuser. Essentially this modifies the overall fan design from more of a shallow ducted propeller to a true tube-axial design.

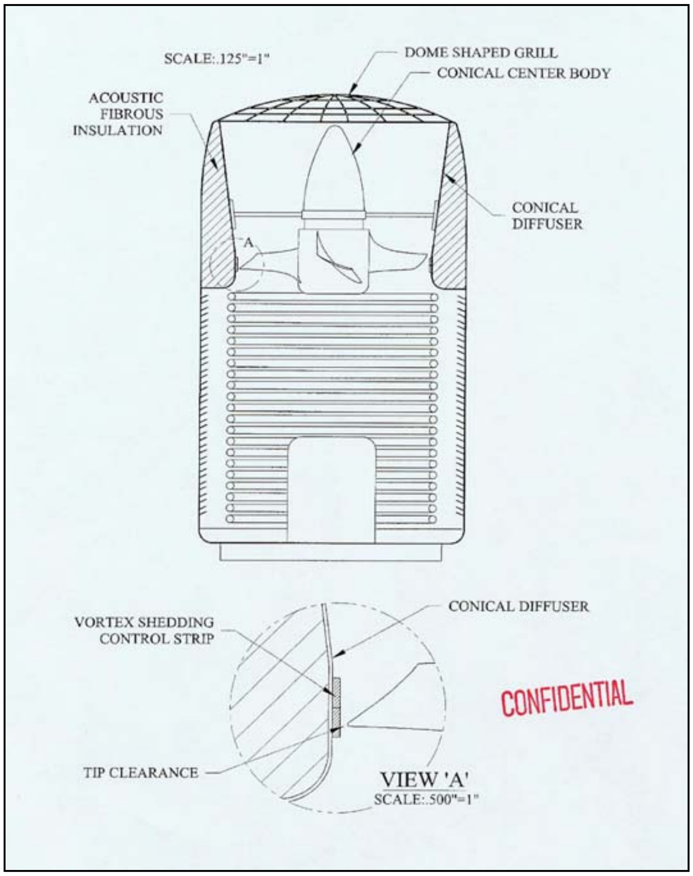

Figure 6. Diagram of the improved condenser fan with enhanced diffuser.

The diffuser increases the exhaust diameter from 19.75" (50 cm) to 24" (61 cm) at the wire grill top. While testing with an experimental stator stage did not demonstrate any added flow, we did find other changes in geometry to yield modest improvement. As the motor occupies the center of the diffuser, the swirl set up by the fan as it expands through the diffuser tends to collapse on the low pressure zone immediately behind the motor. Through trial and error, we found that by using a smooth conical center body on the other side of the motor, flow increased by 20 cfm (9 L/s) and power was reduced by about 2-5 Watts.

Reducing Tip Clearances and Sound Control

The functionality of an air conditioner condenser exhaust is essentially analogous to a ducted fan. Research done over the last twenty years within aeronautical engineering has shown that tip clearance of ducted fan blades to diffuser walls is critical to good performance (Rajagopalan and Zhang, 1989; Abrego and Bulaga, 2002).

Unfortunately, low tip clearances are difficult to manufacture due to required tolerances. Should fan blades strike a solid diffuser wall, the fan blades or motor may be damaged or unacceptable noise created. Thus, in air conditioner fan manufacturing, the fan blades typically have a gap of 0.3 to 0.4 inches (0.8-1.0 cm) to the steel sidewall diffuser. This large tip clearance has a disadvantageous impact on the ducted fan's performance.

Considering the desirability of low sound levels for AC condensers we examined interesting work done at NASA Langley Research Center showing how porous-tipped fan blades in jet turbofan engines can provide sound control by reducing vortex shedding – a known factor in the propagation of excessive fan noise (Khorrami et al. 2002).

Based on the research, it was postulated that rather than using porous fan tips, a porous diffuser sidewall could achieve the same result. This was done by obtaining commercially available 3/16" (0.5 cm) open cell plastic foam 1½" (3.8 cm) wide, and applying it to the inner wall of the diffuser assembly swept by the fan blades. In actual application a UV stabilized open-cell neoprene foam would likely be used. As shown in Figure 6, the foam is applied within the diffuser assembly over the swept blade region to breakup fan blade tip vortices and to reduced sound. We also used a solid tip clearance strip to test the differences. Whereas the solid strip increased fan noise, the open cell foam strip reduced noise markedly (see Table 1).

One added advantage was that the foam can be used to produce very close tip clearances in ducted fans with no danger to the moving blades. Any contact with the foam inner liner will be quickly worn away to yield ideal fan tip clearances. This was verified in overnight tests where tolerances were exceeded.[5] A final advantage is simplicity and cost effectiveness. This is a simple change that can potentially produce large improvements in acoustic and air moving performance.

We estimated the flow and sound impacts of the invention by carefully measuring performance of two fans. Sound levels from fan only operations were measured using hand-held dB meters at the prescribed distance used for *ARI 270-1995* for the horizontal measurements. The results in Table 1 show a large improvement in airflow as well as sound advantages of 2-3 db (15-20% reduction in sound level).

[5] One potential limitation for this approach might be with heat pump applications where freezing rain could potentially build up on the cabinet and the foam, bridging the gap to the fan in along-off-cycle, potentially stalling the motor. However, realistic testing can determine feasible tolerances.

Table 1
Impact on Performance of Reduced Tip Clearance Using Foam Sound Control Strips

Case	**Flow**	**Power**	**Normalized CFM/W**	**dBA**
OEM Fan with slotted grill (1000 rpm) with standard diffuser and top				
Original Configuration	2200 cfm	190 W	11.6	63.0
A5 Fan with 8-pole motor (850 rpm) with extended conical diffuser				
As is (~1/4" clearance)	2110 cfm	135 W	15.6	62.0
Tip clearance < 1/32" foam	2300 cfm	141 W	16.3	60.0
A Fan with 6-pole motor (1100 rpm) with conical diffuser				
As is (~1/4" clearance)	2400 cfm	139 W	17.3	64.5
Tip clearance < 1/32" foam	2610 cfm	145 W	18.0	61.0

Note the improvements in air moving efficiency. As the shaft power requirement increases between the square and the cube of the flow quantity, the advantage of the foam strip for Fan A5 represents a measured improvement in the air moving efficiency of nearly 23%. Moreover, at three feet (1m) away from the condenser, we measured sound reductions of approximately two decibels (15% more quiet to the human ear).[6] In contrast, we had previously attempted a number of other suggested improvements (forward swept blades, dimpled air foils and winglets) which did not produce any measurable sound reduction.

Tests Results

The fans were evaluated with the standard slotted grill top and later with the improved diffuser configuration (elongated diffuser with foam tip clearance strip and conical center body). We also tested with differing PSC motors (six vs. eight pole operating at 1075 and 850 rpm, respectively). The process involved measuring performance and then evaluating the changes in a comparative manner to isolate the best options. While our testing revealed significant performance advantages for the twisted air foil designs, we found that the simple metal blades were fairly well designed given their simplicity. The best performing configurations with and without the enhanced exhaust are summarized below. Table 2 provides numeric data and Figure 7 shows the graphical results.

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[6] Sound measurements were not consistent with the *ARI 270-1995* protocol as we only measured the horizontal sound pressure levels (rather than those overhead) which will tend to accentuate the impact of the expanded diffuser as sound is broadcast upwards rather than outward. However, these measurements may be more indicative of sound nuisance in many residential applications. Numbers are fan only noise levels.

Table 2
Comparative Performance of Fans & Diffuser Elements

	Top	Fan	Motor	Flow (cfm)	Power (W)	CFM/W	Sound (dBA)
*	Slotted	OEM	6-pole	2200	190	11.2	63.0
	Slotted	D	6-pole	2190	150	14.6	65.0
	Slotted	A5	8-pole	1660	130	12.8	62.0
	Diffuser	OEM	6-pole	2250	173	13.0	63.0
**	Diffuser	A5	8-pole	2300	141	16.3	59.0
	Diffuser	A	8-pole	1930	111	17.4	58.0
	Diffuser	A	6-pole	2610	145	18.0	66.0
	Diffuser	E	6-pole	2500	132	18.9	65.0
	Diffuser	E	8-pole	1825	109	16.7	61.0
	Diffuser	D	6-pole	2590	150	17.3	66.0
***	Wire-foam	OEM	6-pole	2250	188	12.0	62.0
	Wire-foam	A5	8-pole	2110	146	14.5	60.0

* Standard configuration, metal bladed fan
** Preferred configuration, with advanced fan and diffuser with foam tip clearance strip
*** Preferred configuration with standard top, wire grill and foam control strip

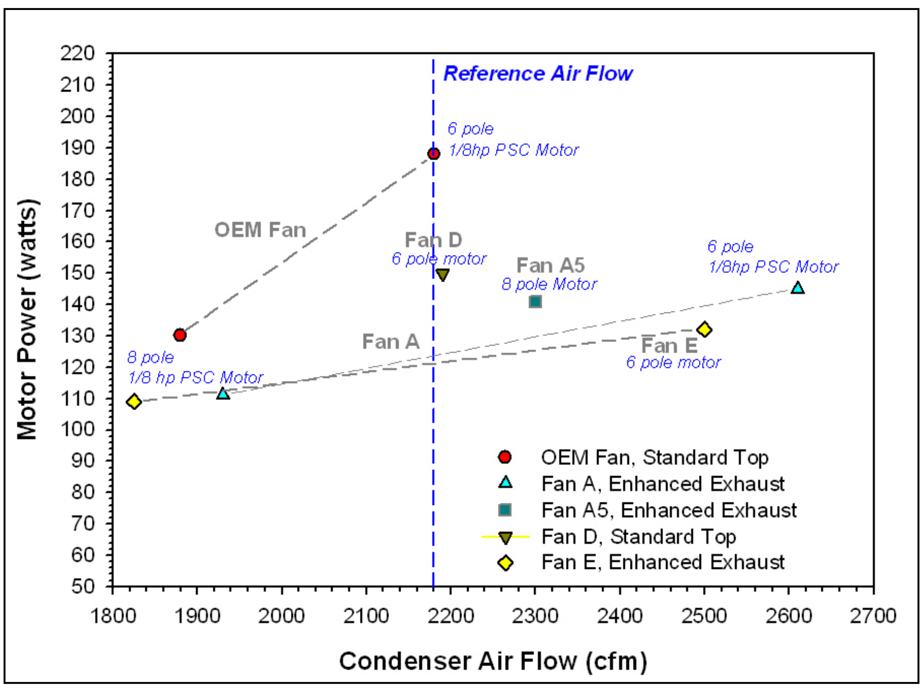

Figure 7. Performance of different fans with and without the enhanced diffuser against the original configuration

Results are shown for each fan, both with the slower 8-pole motor (850 rpm) and the faster 6-pole motor (~1,075 rpm). The best performing fan with the standard slotted grill top and short diffuser was the three-bladed Fan D which was designed for a higher pressure rise. It produces the same flow as the standard OEM fan (~2,200 cfm or 1,038 L/s) with a power consumption of 150 W vs. the 190 Watts for the standard fan at 208 Volts. Sound levels are similar to the standard configuration.

We also did a test of the original configuration with a wire grill rather than a slotted top above the short diffuser. The wire grill showed superior performance– likely due to a lower pressure rise above the fan. The power of the original metal-bladed fan dropped by 2 Watts with equivalent flow. When the foam tip clearance strip was added to the original short diffuser, flow increased by 50 cfm (24 L/s) along with a small drop in sound level.

Also shown are tests using the enhanced diffuser with Fan A, Fan A5 (a 5-bladed asymmetrical version of Fan A) and Fan E. Two fans (B and C) are not shown as results were not promising.

Note that the A5 fan with the diffuser improves air moving efficiency (CFM/W or L/s/W) by greater than 46%. With the enhanced diffuser, this was our preferred configuration. It produced 100 cfm (47 L/s) more flow than the standard configuration while still reducing power by 49 Watts. Unlike Fan D, it also reduces sound since the enhanced diffuser allows the use of a slower turning fan with an 8-pole motor. Fans A and E allow superior flow over the original configuration at even lower power, although sound levels are increased. These results suggest that a fan such as A5 could be run at even lower RPM using a variable speed motor to provide greater power and sound reductions or conversely run at higher RPM to provide greater flows.

Tests with a Variable Speed Motor

To examine how the new fan and diffuser designs would compare with the standard design at a given flow, we obtained a brushless direct current (BDC) 1/3 hp (0.25 kW) motor from a leading U.S. manufacturer. It was programmed to be variable in speed from 0 - 1200 rpm in response to a pulse-width modulated DC signal. These motors tend to be more efficient than PSC motors– by approximately 15% at full speed, but with larger differences at lower speeds. Our tests verified these expectations and also showed the full advantages of the air foil fan designs as well as the enhanced diffuser when flow was equivalent to the OEM design.

For instance, the OEM fan with the standard top and 1/8 hp (0.09 kW) PSC motor required 190 Watts to provide 2,200 cfm (1,038 L/s) of flow at 208 Volts. The original fan with the enhanced diffuser and the BDC motor required 125 Watts to provide the same flow. Thus, the diffuser and the BDC motor produced a power savings of 65 Watts or 34%. However, Fan A5 with the enhanced diffuser, and BDC motor, only required 86 Watts to provide the same flow – a reduction in power of 102 Watts or 55%. The same tests also showed that by increasing rpm with the A5 fan up to 1,060 rpm, flow was increased to 2,410 cfm (1,137 L/s) with 106 W power draw.

Within our research, we established that modulating outdoor unit fan speeds may have attractive performance and sound tradeoffs. We speculate that much of the nuisance of air conditioner noise comes during nighttime when ambient sound levels are low and occupants are asleep. This suggests that the BDC condenser fan speed might be modulated to high speed during very hot daytime periods (e.g. fan speed is high above 94ºF (34ºC) while a low speed would be used when the outdoor temperature was less than 84ºF (29ºC). This would substantially reduce fan noise during evening hours while preserving best peak performance during hot afternoons. Figure 8 shows the comparative performance of the OEM configuration, the OEM fan used with the enhanced diffuser, and BDC motor with the improved fan design.

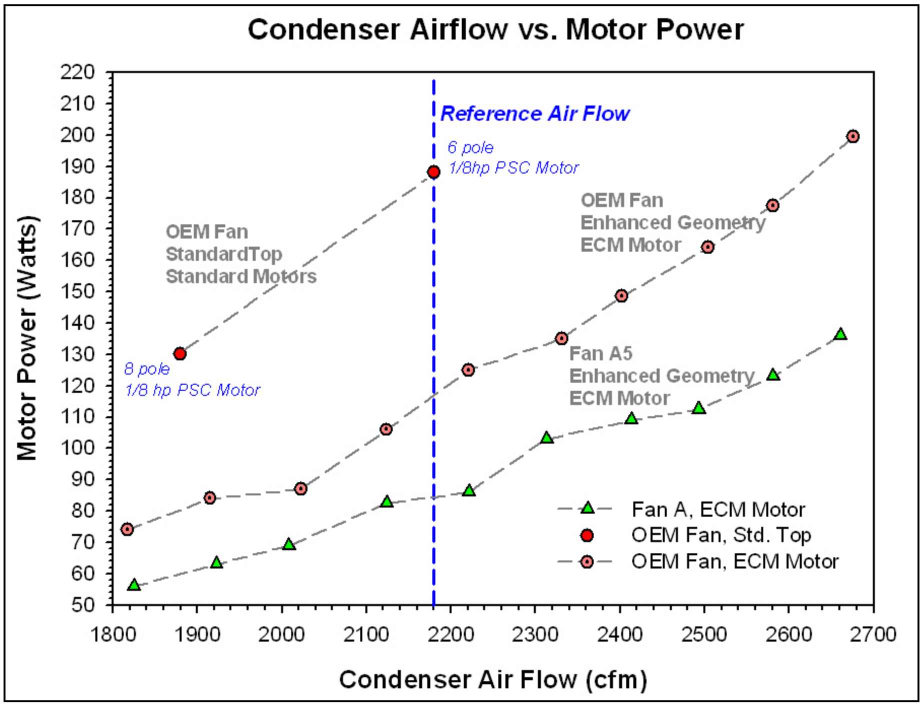

Figure 8. Impact of BDC motor and improved fan blades on condenser performance.

In Figure 8, the impact of the diffuser and BDC motor can be seen as well as the influence of the improved fan blades. Finally, Figure 9 below isolates the airflow improvement created by the tip clearance and sound control foam strip when evaluated at differing flow points with the BDC motor and the high performance fan.

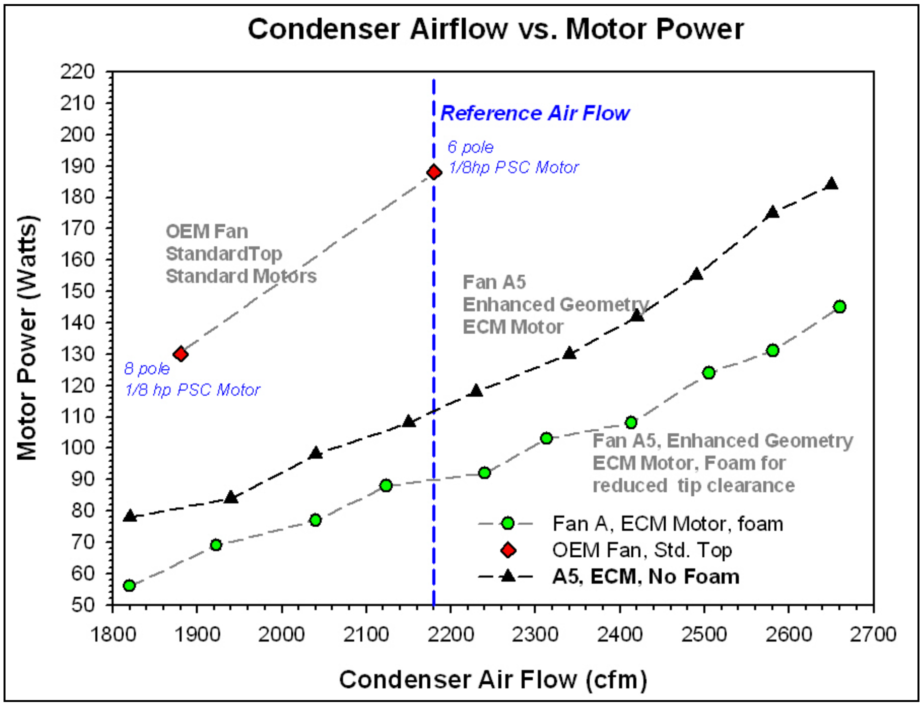

Figure 9. Impact on performance of Fan A5 used with elongated diffuser with and without tip clearance foam strip.

Additional Laboratory Measurements

In May 2004, we took the prototype condenser top to a major U.S. air conditioning manufacturer who possessed a sound room and air flow measuring facilities to allow verification of measurements. Within their laboratories, we found that measured sound reductions of the new diffuser top were only 1-2 dB as measured according to the *ARI 270-1995* standard. This is likely due to the tendency of the longer diffuser to broadcast sound upward to the overhead microphones. We also were able to verify the flow and power reductions previously measured. Power consumption of the PSC motors, when tested at 230 volts was about 10 watts more than tests done at 208 volts. However, the savings with BDC motors should be essentially equivalent at given flows. Also, the savings with PSC motors could be made similar to what we measured by slightly reducing the shaft power of the motors for use with the improved blades. This would improve motor efficiency as the motors approach synchronous speed at higher voltage.

Conclusions

A research project designed, fabricated and tested high efficiency air conditioner condenser fans to improve flow characteristics while decreasing motor power. We also examined potential changes to the condenser exhaust configuration to enhance performance. The primary objective was to reduce motor power while providing similar or superior flow to the standard configuration. A secondary objective was to provide sound reductions which is important to consumers.

Within the effort, we developed new twisted and tapered propeller air foils that demonstrated greater air moving efficiency. Fan-only savings were 40 watts (21%) for the same motor and condenser top.

We also showed how a lengthened diffuser with a conical insert after the motor can improve air moving efficiency by over 16% for standard fans and over 27% for high performance fans. Fan tip losses and associated vortex shedding was reduced though the use of a porous foam strip to improve air flow performance while helping to reduce sound. This also allows slower fans to provide superior air moving performance along with sound control. On the negative side, however such design would lengthen condenser height by approximately 18” (46 cm) and somewhat increase costs by increasing sheet metal requirements.

The fundamental project achievements are summarized below:

- Provides 49 Watt reduction in fan power (141 W vs. 190 Watts) with PSC motors at 208 volts.

- Increases condenser air flow by 100 cfm or 47 L/s (5% increase in fan flow).

- Provides 102 W power reduction with BDC motor.

- Reduces fan-only ambient sound level by 1-2 dBA. Ground level sound reduction is greater.

- BDC motor allows lower fan speeds for ultra-quiet night operation, higher flows for maximum capacity during very hot periods (temperature based control)

<u>Key Technologies</u>

- High efficiency 5-bladed asymmetrical fan moves air quietly at lower fan speeds.

- Diffuser top for effective pressure recovery allowing increased air flow at low speeds.

- Conical center body reduces losses in exhaust swirl.

- Foam sound control strip to reduce tip losses and fan tip vortex shedding.

- Patents pending: U.S. Application Serial No. 10/400,888, Provisional applications 60/369,050 / 60/438,035 & UCF-449CIP; *WhisperGuard* (UCF-Docket No. UCF-458).

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References

A. I. Abrego and R. W. Bulaga, "Performance Study of a Ducted Fan System," NASA Ames Research Center, American Helicopter Society Aerodynamics, Acoustics and Test Evaluation Technical Specialists Meeting," San Francisco, CA, January 23-25, 2002.

ANSI/ASHRAE 51-1985 (Revision: ANSI/ASHRAE 51/AMCA 210-1999): Laboratory Methods of Testing Fans for Rating, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

ARI, 2003. "Standard 210/240: Unitary Air Conditioning and Air Source Heat Pump Equipment," Air Conditioning and Refrigeration Institute, Arlington, VA.

ARI, 1995. ARI 270-1995: Sound Rating of Outdoor Unitary Air Conditioning Equipment, Air Conditioning and Refrigeration Institute, Arlington, VA.

ASHRAE, 2004, "Chapter 18: Fans," HVAC Systems and Equipment, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.

R. D. Blevins, Applied Fluid Dynamics Handbook, "Nozzles, Venturis and Diffusers," Van Nostrand Reinhold, NY, 1984.

J. S. Bradley, 1993, "Noise from Air Conditioners," Institute for Research in Construction, National Research Council of Canada, Ottawa, ONT.

DOE/EIA, 1999. A Look at Residential Energy Consumption in 1997, Energy InformationAdministration, DOE/EIA-0632 (97), Washington, DC.

S. K. Fischer and C. K. Rice, 1983. The Oak Ridge Heat Pump Models: I. A Steady-State Computer Design Model for Air-to-Air Heat Pumps, ORNL/CON-80/R1, Oak Ridge National Laboratories, August, 1983

R. H. Green and L. Roberts, 1996. "The Effect of Air-Coil Design on the Performance of Heat Pumps and Air Conditioners," ASHRAE Transactions, Vol. 102, Part 1, pp.257-265.

D. Japikse and N.C. Blaines, 1993. Diffuser Design Technology, Concepts ETI, Inc., White River, VT.

M. R. Khorrami, F. Li and M. Choudhari, 2001. “A Novel Approach for Reducing Rotor Tip Clearance Induced Noise in Turbofan Engines” NASA Langley Research Center, American Institute of Aeronautics and Astronautics, 7th AIAA/CEAS Aeroacoustics Conference, Maastrictht, Netherlands, 28-30 May, 2001.

Nicholas C. Kernstock, “Slashing through the Noise Barrier,” Aviation Today, August 1999.

W.P. Levins, C.K. Rice, and V.D. Baxter, 1997. "Modeled and Measured Effects of Compressor Downsizing in an Existing Air Conditioner/Heat Pump in the Cooling Mode," ASHRAE Transactions, Vol. 192, Part 2, pp.22-33.

J. Proctor, Z. Katsnelson, G. Peterson and A. Edminster, Investigation of Peak Electric Load Impacts of High SEER Residential HVAC Units, Pacific Gas and Electric Company, San Francisco, CA., September, 1994.

R. G. Rajagopalan and Z. Zhang, "Performance and Flow Field of a Ducted Propeller," American Institute of Aeronautics and Astronautics, 25th Joint Propulsion Conference, AIAA-89-2673, July 1989.

G. J. Rosenquist, 1997. "Comparison of Simulated and Measured Test Data on Air-Source Heat Pumps", 3rd International Conference on Heat Pumps in Cold Climates, Wolfville, Nova Scotia, Canada, Aug. 11-12, 1997; Caneta Research, Inc., Mississauga, Ontario, Canada, November, pp.391-405.

M. Schaffer, 1991. A Practical Guide to Noise and Vibration Control for HVAC Systems, ASHRAE, Atlanta, GA.