FIELD MEASUREMENT OF SOUND FROM RESIDENTIAL VENTILATION FANS

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FROM RESIDENTIAL
VENTILATION FANS

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Institut de recherche en construction

CLIENT REPORT

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Field Measurement of Sound from Residential Ventilation Fans

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1. Introduction

A CSA standard was developed for laboratory testing of residential ventilation fans to provide air-handling and sound emission ratings. A study was performed to evaluate these laboratory test methods, and to assess the relationship between the laboratory ratings and actual field performance. The study was commissioned by a consortium of interested parties, including the Research Division of Canada Mortgage and Housing Corporation (CMHC) - the direct client for the work reported here - and Ontario Hydro.

The first phase of that study was laboratory testing of 11 ventilation units to rate both airflow and sound power emission according to the current draft of CSA C260; the results were presented in report CR5899-1.

The second phase was field testing of these same units after installation in residences. The five bathroom ceiling exhaust fans and five kitchen rangehood fans were to be supplied to a local (Ottawa area) contractor, who would install them in new houses. Due to a shortage of cooperative contractors, and a slowdown in building starts, the actual field study included only six of the intended ten fan tests.

Both the airflow and noise parts of the field study are dealt with in this report. The flow testing is treated briefly, due to its simplicity. The acoustics tests were planned not just to provide sound power data for comparison with the laboratory results, but also to assess the relationship between the sound power and the resulting sound pressure levels observed in the rooms. It is the sound pressure level in situ that determines user annoyance.

The main body of the report presents an overview of the work performed, and a discussion of the significant issues. This presentation is divided into five parts:

Part 2: installation details and measurement procedures;
Part 3: air flow results;
Part 4: sound power results;
Part 5: sound pressure level results; and
Part 6: summary.

The summary is followed by appendices to tabulate data for each fan.

2. Installation Environment

The fans were installed in the kitchens and bathrooms of a series of new two-storey semi-detached houses constructed by Maisons Enertek in Gatineau, Québec. All fans are identified by the numbers on labels applied in the first stage of testing - the laboratory airflow measurements at Ortech International.

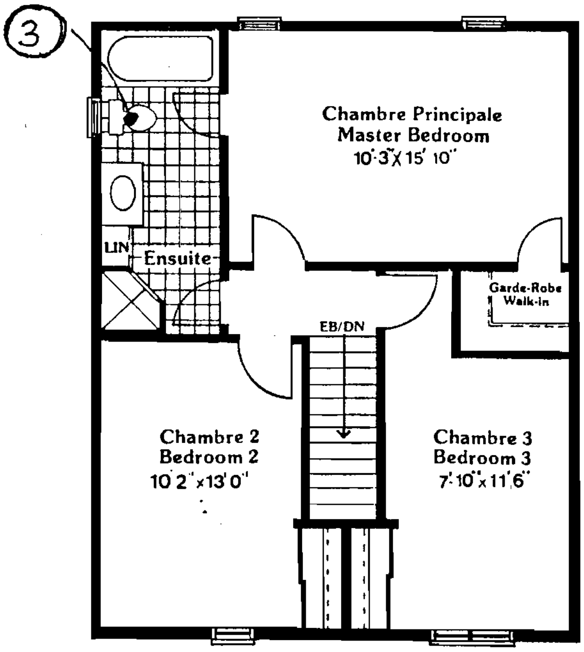
Each of these houses had its main bathroom on the upper floor, and a kitchen and smaller bathroom on the lower floor. Figure 1 presents the floor plans; the locations of the ventilation units being tested are indicated on the plans. The details of fan installation are given in Table 1.

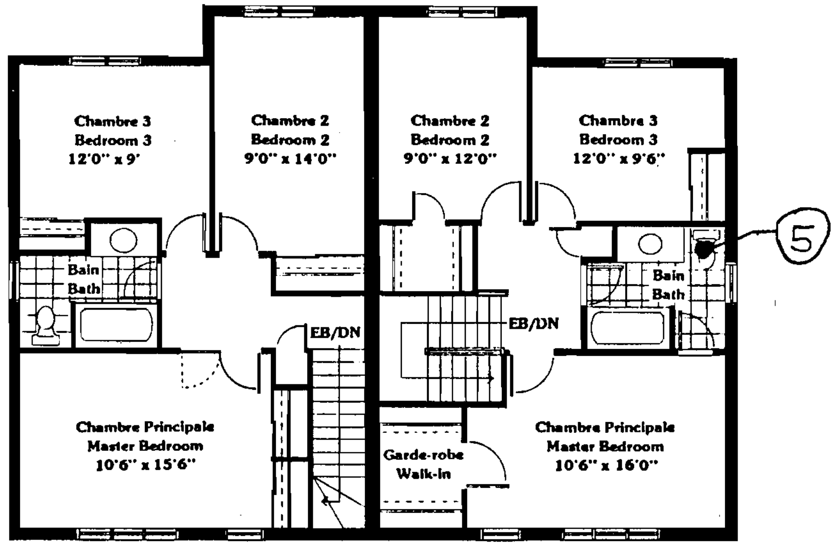
Table 1: Fan locations for field tests of units previously measured according to CSA C260, and air flow measured in field tests.

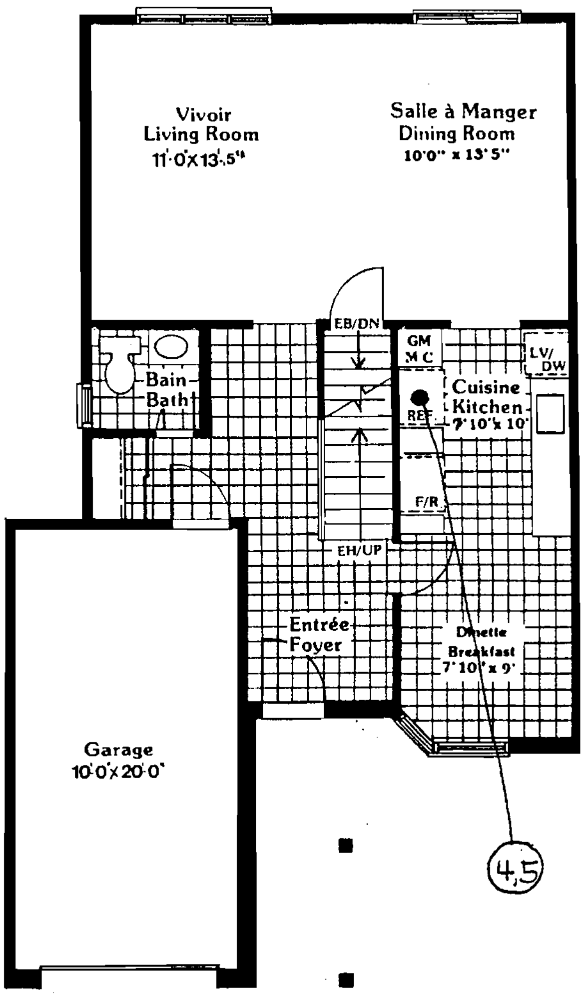
Fan	Rise (m)	Run (m)	Type of ducting	Flow (L/s)
Bathroom #1 (first floor)	-2.4	2.5	flex corners galvanized runs	28
Bathroom #5 (second floor)	-5.0	0.5	100 mm flex to basement galvanized horizontal run	32
Bathroom #3 (second floor)	0	2.0	100 mm flex	28
Kitchen #3	0.5	4.0	150 mm flex	42
Kitchen #5	-1.8	4.4	85 x 250 mm drop to floor 125 galvanized horizontal	78
Kitchen #4	-1.8	4.4	(same as #5)	63

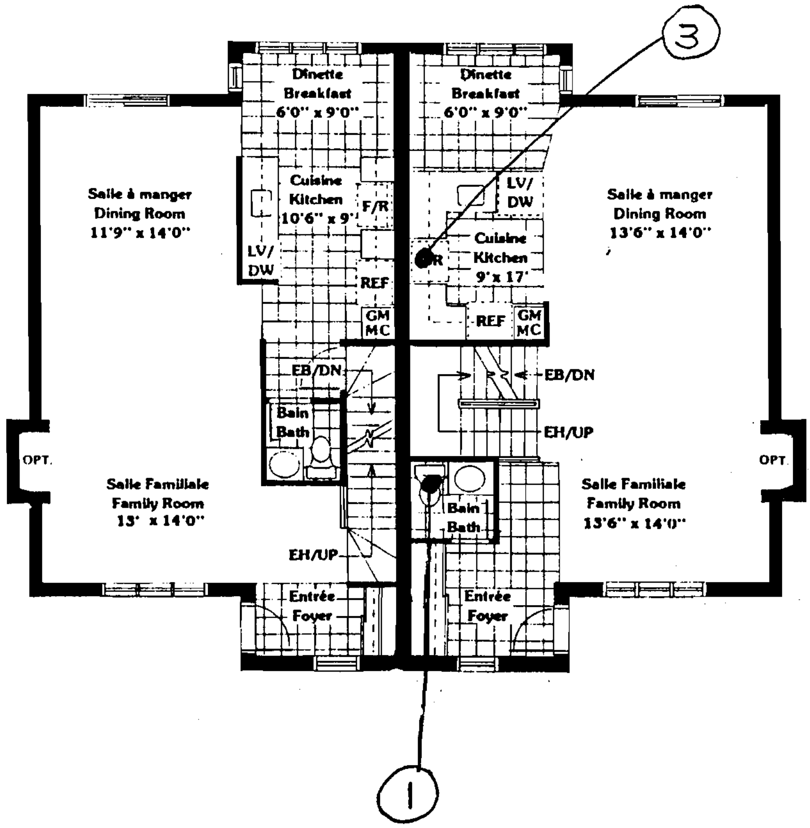
The construction was essentially complete in the test rooms, except for floor tiles in some cases. None of the floors of the kitchens or bathrooms were carpeted, and the rooms were noticeably bare and reverberant. All doors were closed at the time of testing. Closing the bathroom doors provided well-defined rooms. The kitchens, however, had open archways to adjacent rooms, which had some effect on the measured sound pressure levels, as will be discussed further.

For the sound pressure level and related reverberation time measurements, two absorptive panels (about 2 metric Sabins) were added to bring acoustic absorption into the range expected with normal furnishings. Five absorptive panels were added for the acoustic intensity measurements, as discussed on the following pages.


3
Chambre Principale
Master Bedroom
10'3"x15' 10"
LIN
Ensuite
Garde-Robe
Walk-in
EB/DN
Chambre 2
Bedroom 2
10'2"x13'0"
Chambre 3
Bedroom 3
7'10"x11'6"


Chambre 3
Bedroom 3
12'0" x 9'
Chambre 2
Bedroom 2
9'0" x 14'0"
Chambre 2
Bedroom 2
9'0" x 12'0"
Chambre 3
Bedroom 3
12'0" x 9'6"
5
Bain
Bath
Bain
Bath
EB/DN
EB/DN
Chambre Principale
Master Bedroom
10'6" x 15'6"
Garde-robe
Walk-in
Chambre Principale
Master Bedroom
10'6" x 16'0"


Vivoir
Living Room
11'0"x13'5"
Salle à Manger
Dining Room
10'0" x 13'5"
EB/DN
GM
MC
LV/
DW
Bain
Bath
Cuisine
Kitchen
7'10"x10'
REF
F/R
EH/UP
Entrée
Foyer
Dinette
Breakfast
7'10"x 9'
Garage
10'0"x20'0"
4,5


Dinette
Breakfast
6'0" x 9'0"
Dinette
Breakfast
6'0" x 9'0"
3
Salle à manger
Dining Room
11'9" x 14'0"
Cuisine
Kitchen
10'6" x 9'
F/R
LV/
DW
REF
GM
MC
LV/
DW
Cuisine
Kitchen
9' x 17'
REF
GM
MC
Salle à manger
Dining Room
13'6" x 14'0"
EB/DN
EB/DN
EH/UP
Bain
Bath
Bain
Bath
OPT.
OPT.
Salle Familiale
Family Room
13' x 14'0"
Salle Familiale
Family Room
13'6" x 14'0"
EH/UP
Entrée
Foyer
Entrée
Foyer
1

2.2 Air Flow Measurement

The exhaust airflow was measured by D. Fugler of CMHC, using a research device developed for CMHC in 1988, as described in the CMHC Research Division report "Duct Test Rig". This portable unit measures airflow by the pressure drop across a series of calibrated orifices. It uses an internal fan to match the exhaust airflow, thereby minimizing the test rig's effect on the exhaust fan inlet conditions. The flows are matched when there is neutral static pressure in the enclosed space between the orifices and the fan to be measured.

For the bathroom fans, the duct test rig was simply held against the ceiling, with a foam gasket sealing between the test rig and the ceiling surface around the grille of the exhaust fan. For the kitchen fans, a sheet of polyethylene was taped to the rangehood on one end and the test rig casing at the other end. Curiously, this plastic sheet provided a visual check on the test rig instrumentation, as it would visibly slacken when the pressure null was attained.

Note that what was being measured was the exhaust flow at the fan itself. Downstream leakage from the ducts into the indoor air would reduce the actual amount of air exhausted. Testing at the exterior outlet would resolve this, but the effects of outdoor wind on the instrumentation and the inconvenient exhaust outlet locations (at grade, in soffits, etc.) made such testing impractical. Further, it was the flow *for the fan itself* which had previously been assessed in the laboratory testing.

2.3 Sound Power Measurement

All the acoustical measurements were made in one-third-octave bands using a Norwegian Electronics NE-830 acoustical analyzer. Detailed data were stored on diskette for subsequent analysis; A-weighted results were checked throughout the measurement process, as a preliminary quality control, and any questionable measurements were repeated.

A Norwegian Electronics Model 216 acoustic intensity probe was used for the sound power measurements. Acoustic intensity is determined from a series of measurements distributed over a "surface" enclosing the source. A set of plane surfaces enclosing the sound source was selected for each specimen, and the edges of these surfaces were marked clearly. Pieces of string attached to the adjacent cupboards were used to mark the surface limits in the case of the kitchen rangehoods, and a moveable stand was used for the bathroom fans. For the rangehoods, six such surfaces were needed, whereas three or four surfaces of a simple rectangular box were used for the bathroom fans (three surfaces if the fan was near a room corner, four surfaces if the fan was adjacent to only one wall). For each of the plane surfaces, the acoustic intensity for that surface was determined by a 30-second scan with the intensity probe, moving the probe to sample the whole surface area as uniformly as possible. The probe was oriented to measure intensity perpendicular to the surface, and each scan was repeated with the probe direction reversed (which should give the same intensity level but reversed direction). If the overall A-weighted acoustic intensity magnitudes for the two probe directions did not agree within 0.5 dBA, the measurement was repeated. In most cases, agreement was within 0.2 dBA or better.

The sound power from each measurement plane was calculated from the measured intensity and the surface area, and these partial sound powers were combined to give the overall sound power level.

No standard procedures for assessing precision or validity of scanned acoustic intensity measurements have been developed yet, and obtaining reliable acoustic intensity measurements in a reverberant space is not trivial. Several precautions were taken to provide better measurement conditions and permit subsequent evaluation of measurement quality. A set of five absorptive panels (each providing about 1 metric Sabin of absorption) were placed nearby during the acoustic intensity measurements, to reduce the reverberant sound. The equivalent sound pressure level at the probe was also recorded, to permit subsequent checks that the reactivity index (the difference between sound pressure level and intensity level) was within acceptable limits. Repeating each measurement with the probe reversed is the most reliable way to check measurement validity, and this was done for each measurement surface.

2.4 Sound Pressure Level Measurements

The sound pressure level measurements were made in one-third-octave bands using a Norwegian Electronics NE-830 analyzer with a 13 mm diameter condenser microphone (B&K 4165 on type 2619 preamplifier). The microphone sensitivity was checked with a Bruel & Kjaer calibrator at the beginning and end of measurements on each fan, to ensure accurate calibration; no significant changes in calibration were observed.

The purpose of the sound pressure level measurement was to determine the sound pressure level versus distance from each ventilation device. These measurements were made in three directions, at set distances (0.5, 0.7, 1.0, 1.4, and 2.0 metres) from the centre of each fan unit. These provide a measure of the sound levels that users would experience in these rooms when the fans are in operation.

As noted previously, two absorptive panels (providing about 2 metric Sabins of absorption) were added to the rooms for these measurements to approximate the effect of normal furnishings.

Acoustic absorption in each room (including the two added panels) was determined from measured sound decay rates, using the NE-830 analyzer with a Tracoustics NS-100 amplifier/loudspeaker system as the sound source. Decay measurements were made at three or more positions in each room; for each position, the ensemble average of four decays was measured.

3. Air Flow Measurement Results

The results of the air flow measurements are shown in Table 1. The three bathroom fans were quite consistent, with flows of about 30 L/s. The kitchen fans ranged from 42 to 78 L/s.

The corresponding laboratory results were approximately 48 and 91 L/s respectively. Flows measured according to CSA C260 are at a static pressure of 25 Pa. Although the field results are small in number, the consistently lower flows suggest that a more realistic estimate of installed airflow could be obtained using a static pressure of 50 to 75 Pa. The causes of the higher flow resistance include the use of flex ducting, elbows, long duct runs, etc. in the field installations. Kitchen fan #3 suffered reduced flow because the sheet metal exhaust hood on the outside wall had been badly creased, making it impossible for the backdraft damper to open fully when the fan was in operation.

4. Sound Power Results

The detailed sound power results for each ventilation device are presented in the Appendix, together with the corresponding laboratory data from measurements according to CSA C260.

Agreement between the laboratory and field data was fairly close. The rangehoods gave slightly higher sound power levels in the field measurements, but the bathroom exhaust fans had lower sound power in the field. The average deviation in the A-weighted sound power was was 1.4 dB.

Too few measurements were made for a statistically meaningful assessment of precision of the field sound power data, but repeating the intensity measurements with probe direction reversed gave a rough indication. For the frequency bands from 250 Hz to 2500 Hz (which dominate the A-weighted sound power), the intensity results typically agreed within 0.5 dB or better, and the reactivity index was acceptable (less than 10 dB). At frequencies from 100 Hz to 200 Hz and above 2500 Hz, intensity variations from 0.5 to 1 dB were typical. This would suggest repeatabilty for the overall A-weighted sound power level of about 1 dB. In addition, there might be systematic bias in these measurements, as discussed below.

Although the observed differences between laboratory and field results were generally comparable to the combined experimental uncertainty, some were considered sufficiently consistent or prominent to warrant discussion. Some consistent trends are evident for the bathroom exhaust fans; as an example, the laboratory and field sound power results for bathroom exhaust fan #3 are presented in Figure 2.

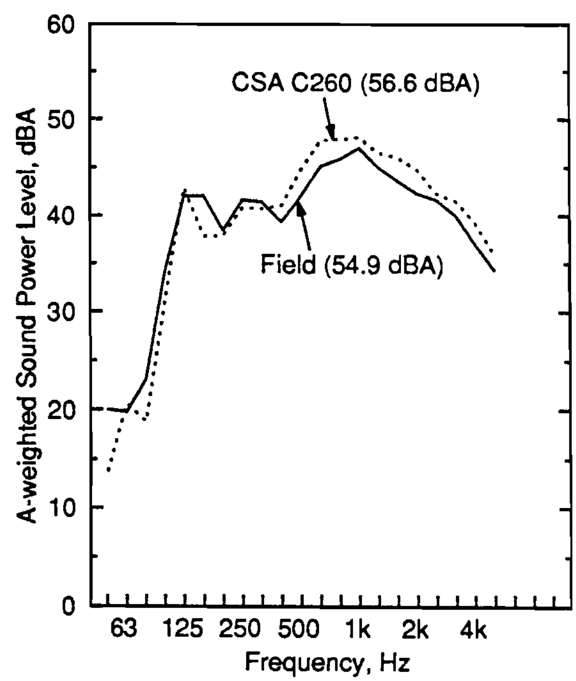


Figure 2: Laboratory and field sound power results for bathroom exhaust fan #3.

The two curves in Figure 2 are very similar. At frequencies above 400 Hz, the field sound power is lower than the laboratory result; a similar trend is evident for all the bathroom fans. Consistency of the trend in a frequency range for which measurement precision should be best suggests this is a systematic effect. This accounts for the lower A-weighted sound power from the bathroom fans in the field test.

Such a reduction is presumably indicative of a difference in fan operation or a systematic difference between the two sets of measurements.

Unfortunately, the fan speed and the electrical voltage were not measured during the field tests. However, a voltage check on a subsequent visit matched the laboratory condition (120 V) within measurement precision. The lower air flow observed in the field suggests a higher static pressure in the field installations, but this would tend to increase fan speed and hence increase the sound power emission (as shown in Fig. 10 of report CR-5899.1). Thus there is no direct evidence of changed operating conditions to explain the observed reduction in sound power emission of the bathroom exhaust fans.

The most likely explanation is systematic error in the field measurement of sound power by acoustic intensity. The sensitivity of any acoustic intensity probe varies with frequency and with directionality of the sound field. Careful comparison of sound power measurements with a small source in the IRC reverberation room, versus acoustic intensity measurements with several intensity probes in the IRC anechoic room, suggest a slight decrease in sensitivity of the Norwegian Electronics 216 acoustic intensity probe above 1 kHz. The effect was very similar to the difference between laboratory and field sound power results for these fans. In the ideal case with near plane-wave propagation in the anechoic room, the theoretical deviation from uniform sensitivity can be calculated, but for the much more complex fields in these bathrooms and kitchens this is not feasible. The results should simply be viewed as indicative of characteristic limitations of the acoustic intensity technique.

In the case of the rangehood fans, the overall sound power emission is greater in the field measurements. The change may be due to different installation - the fans were tested with vertical discharge in the laboratory, but were installed with horizontal discharge in the field. The manufacturer's rating is 5.5 sones for vertical discharge, and 6.0 sones for horizontal discharge, so an increase between 1 and 2 dB would be expected in the A-weighted level.

The data for rangehood #5 are given in Figure 3. The sound power in the 500 Hz to 2 kHz range does not exhibit a consistent change from laboratory to field results. Most of the increase in overall sound power in the field results is due to a strong 125 Hz tone evident in the field results for rangehood fans #4 and #5.

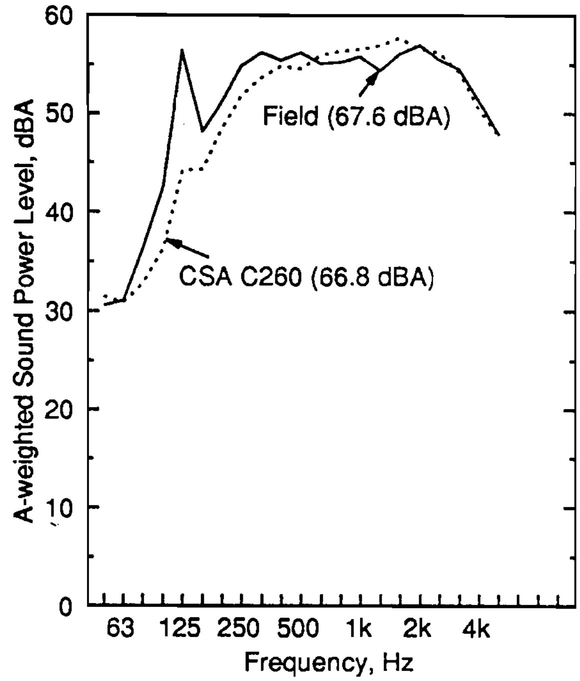


Figure 3: Laboratory and field sound power results for kitchen rangehood fan #5.

No substantial tone was evident in the laboratory results (or in the field result for rangehood #3) and there is no obvious explanation for the tones in the field results. It is perhaps significant that these fans failed after a few weeks' use; possibly the tones were an early warning of a problem.

Despite the specific differences discussed above, the typical change between the laboratory and field results of about 1.5 dB in overall A-weighted sound power would be barely detectable as a change in loudness. Agreement between laboratory rating and field performance is certainly good enough for practical purposes.

5. Sound Pressure Level vs Sound Power

Sound power ratings (like light bulb power in watts) relate to the property of a source rather than the intensity observed when the source is installed in a room. It is the resulting sound pressure level in a room that affects an observer's assessment of loudness (just as the observer responds to how bright the room is in the case of the light bulb). This depends not just on the source, but also on the receiver's location and the characteristics of the room. The larger the room and the more acoustically absorptive its surfaces and furnishings, the lower the resulting sound pressure level.

The sound pressure level measurements in this study were performed to establish the relationship between sound power and sound pressure levels in typical bathroom and kitchen environments. Figure 4 shows the A-weighted sound pressure level relative to the sound power level for each fan tested.

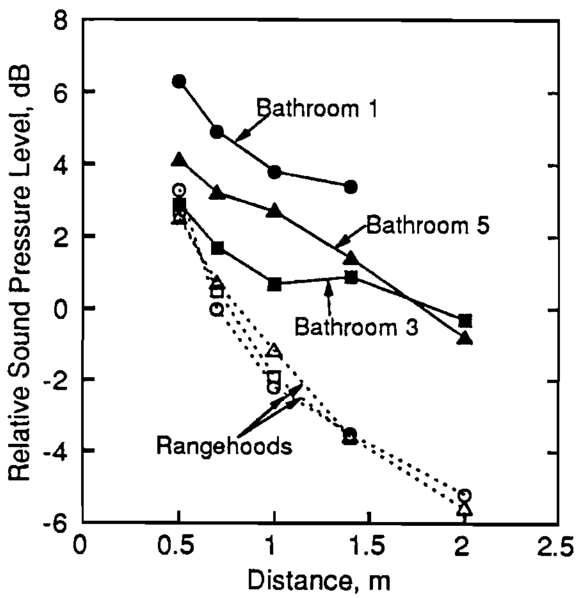


Figure 4: A-weighted sound pressure level relative to sound power level for the fans tested in this study.

Clearly there are substantial differences among the results, but certain trends are obvious:

- for all fans, the sound pressure level decreases with increasing distance from the fan, so perceived loudness depends on typical distance from the source in normal use;
- the difference among the results for the six fans is strongly related to room characteristics, with the highest relative sound pressure level in the smallest room (a tiny bathroom with about 1.5 m^2 floor area) and the lowest levels in the largest rooms (the three kitchens, all about 15 m^2 floor area). This trend can be explained by the effect of acoustic absorption in these rooms.

5.1 Diffuse Field Theory Prediction

The sound pressure level can be predicted quite well, given the acoustic absorption and the location of the sound source in the room. Figure 5 shows the measured sound pressure level for bathroom fan #1 (relative to the field sound power) together with the sound pressure predicted by several calculations.

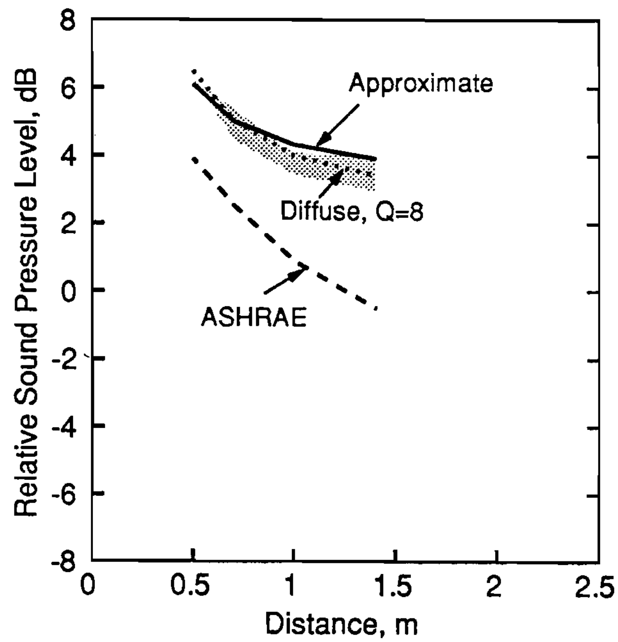


Figure 5: Measured and predicted A-weighted sound pressure level relative to sound power level for ceiling exhaust fan #1. The gray region indicates the range of measured values. Labelled curves give results with three prediction methods.

Traditional diffuse sound field theory predicts the sound pressure level in a reverberant room will exceed the corresponding sound power level by:

$$10 \log [Q/4 \pi r^2 + 4/R] \qquad (1)$$

where:

Q is the source directivity factor,

r is the distance from the source, in metres, and

R is the "room factor", approximately equal to room absorption, in metric Sabins.

The source directivity factor, Q, can depend on the specific source in rather complicated ways, but for simple sources is primarily determined by the number of sound reflecting surfaces adjacent to the source. For a fan near the junction of one wall and the ceiling (two surfaces), Q is about $2^2 = 4$. For a source near a corner (three surfaces), Q is approximately $2^3 = 8$.

Approximating R by the room absorption, the term 4/R can be calculated from the measured sound decay rate in each room using the traditional Sabine expression:

$$\text{Absorption (metric Sabins)} = 0.161\ V / T_{60} \qquad (2)$$

where:

V is the room volume, in m^3, and

T_{60} is the reverberation time, in seconds.

For the case illustrated in Figure 5 (a bathroom exhaust fan near a room corner), Q is 8 and the average measured absorption is 2.1 metric Sabins. The dashed line labelled "Diffuse, Q=8" is the sound pressure level predicted using these values in Eqs. 1-2. Agreement with the diffuse field prediction is excellent in this case, and quite good for the other bathroom fans (shown in graphs in the Appendix).

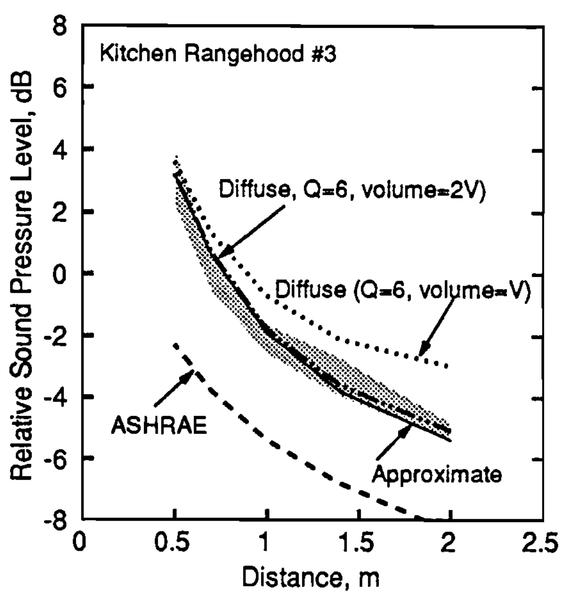


Figure 6: Measured and predicted A-weighted sound pressure level relative to sound power level for kitchen rangehood #3. The gray region indicates the range of measured values. Labelled curves give results with three prediction methods.

Figure 6 shows the measured relative sound pressure level, together with predictions for a rangehood unit. For the rangehood fans, the prediction is more difficult because the open archway from the kitchens to adjacent rooms makes the effective room volume (V in Eq. 2) larger than the actual room size, and complicated cupboard geometry adjacent to the fan makes selection of the appropriate value of Q rather arbitrary. The wall behind the fan and the effect of the overhanging hood ensure Q is greater than 4, but the cupboards are partly open; as a compromise, Q=6 was used. Curves were calculated for V equal to room volume or twice the room volume; the latter gives better agreement with measured values.

Clearly, the prediction can be adjusted to match the measured results, and arguments can be made to justify the values of Q and V selected. However, the calculation requires "knowledgeable application" of the expressions, which could be less charitably described as arbitrary fitting to get any desired answer. Actual results deviate appreciably from the predictions, as evident from some less ideal results in the Appendix. These deviations are presumably because of the local effect of reflections from other room surfaces, and different effects of similar strength could be expected in other rooms. The strong similarity of the three rangehood results is presumably due to the nearly identical adjacent reflecting surfaces (cupboards) and similar room dimensions in these three kitchens.

5.2 ASHRAE Prediction

Figures 5 and 6 each have a dashed line labelled "ASHRAE", calculated using Eq. 15 from Chapter 32 of the ASHRAE Systems Handbook. This expression is intended to predict sound pressure level in typical offices due to the sound power radiated from a ventilating duct outlet. It was derived from a regression analysis of measurements at distances greater than 1 m from sound sources in "typically furnished" rooms.

The ASHRAE expression predicts the sound pressure level in a room will exceed the corresponding sound power level by:

$$-5\log(V) - 3\log(f) - 10\log(r) + 12 \qquad (3)$$

where:

V is the room volume, in m^2;

f is the octave-band center frequency, in Hz; and

r is the distance from the source, in m.

For application in this study, f was taken as 1000 Hz, in the middle of frequency range dominating the sound power.

The ASHRAE expression clearly predicts lower sound pressure levels than were observed, as evident in Figures 5 and 6 and in the corresponding graphs for other fans in the Appendix. This deviation is not surprising, because the source directionality and room absorption conditions in this study are significantly different from the case for which the ASHRAE expessions were derived.

5.2 Approximate Diffuse Field Prediction

A much simpler (if slightly arbitrary) approach is to apply Eq. 1 with a common set of parameters for all the fans.

For all cases, the value of Q fell between 4 and 8; using a value of 6 is a reasonable compromise. Assuming the room factor R is equal to room floor area (in m^2) gave reasonably good agreement with all the data. This seems to be an over-estimate of the actual absorption in typically furnished kitchens or bathrooms - because these rooms normally have rather hard surfaces unlike the carpets and upholstrered furniture in typical living rooms. However, some compensation is needed for sound energy flow through open doors or archways. Although the specific parameters cannot be rigorously justified for any specific case, they do provide a qualitatively sensible estimate.

Results of this prediction are given by curves labelled "Approximate" in Figures 5 and 6, and in corresponding figures for the other fans in Appendix B. For all cases, the "Approximate" curve shows good agreement with the range of sound pressure levels actually observed in each room.

Thus a reasonably simple and accurate method has been determined for predicting the sound level experienced by users of typical residential fans. This requires simply the A-weighted sound power level for the fan (which could be obtained from testing according to CSA C260) and knowledge of the room dimensions. This method could be incorporated in an appendix to CSA C260 at its next revision.

6. Summary

Overall, the sound power measured for the fans in their field installations agreed quite closely with the laboratory results. Deviations from the laboratory results could be due to fan installation or measurement bias, but were in any case within the experimental uncertainty.

The relationship between sound power level and the resulting sound pressure level in the rooms can be explained quite well by simple theory. An approximate version of diffuse field expression can predict sound pressure level in typical kitchens or bathrooms with rather good accuracy.

From a design point of view, this procedure would provide the basis for selecting appropriate fans, if users were given criteria for "acceptable" values of fan noise. As yet, to the author's knowledge, no substantial social survey has been done to establish such criteria. This obviously would warrant further research.

Overall, a satisfactory technical approach for design or regulation of noise from residential ventilation fans appears to be quite clearly established, despite the limited sample in this study.

The author notes that the air flow measurements, and their description in this report are the work of D. Fugler of Canada Mortgage and Housing Corporation. The cooperation of M. Roger Brazeau of Maisons Enertek is also gratefully acknowledged.