Addressing Noise Problems In Screw Chillers

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Two 300 ton (1055 kW), water-cooled rotary screw chillers and a 125 ton (440 kW), air-cooled rotary screw chiller caused noise problems at two different hospitals. The case histories of these chillers demonstrate a noise problem known as "pure tone" noise, which often is overlooked in the design process by the practicing HVAC engineer.

Three reasons for overlooking pure tone noise are: (1) The difficulty of obtaining manufacturers' 1/3 octave-band sound power or sound pressure level data needed to identify the severity of pure tone noise components; (2) The absence of a comprehensive industry noise methodology that can describe fully human subjective response to pure tone noise; and (3) Not understanding how to evaluate manufacturers' 1/3 octave-band data when it is available. Before proceeding with the case histories and describing how to identify pure tone noise sources, a brief discussion regarding pure tones is appropriate.

HVAC systems have many types of noise sources. Examples include airflow through ductwork, air terminals or air devices; mechanical fan noise; and blower, burner or compressor noise from boiler and refrigeration equipment. All of these noise sources are a composite of many discrete and separate frequencies of sound (units of cycles per second, denoted as Hertz [Hz]). When many random frequencies of sound are in the audible range of the human ear, the sound usually is characterized as "broadband." However, if one or more sound frequencies stand out above the adjacent frequencies (usually by more than 5 to 8 decibels [dB]), then the broadband sound is said to contain prominent pure tones.

Examples of pure tones in HVAC systems include multiples of the blade passing frequency of a fan (number of blades times the revolutions per minute [rpm] divided by 60), the low hum from a hermetic compressor (number of cylinders × rpm /60), or the annoying tone from a centrifugal or screw compressor (number of compressor blades or rotor lobes × rpm /60).

Pure tone noises are much more annoying to the human ear than random broadband noise. Given equal dB levels of noise from random broadband and pure tone noise sources, the pure tone noise is more likely to cause a noise complaint.

The rotary screw chillers described in the following case histories have double helical rotary screw compressors with the male rotor operating at a nominal 3,600 rpm. Case History 1 describes the pure tone noise problems associated with two open-drive, 300 ton (1055 kW), water-cooled rotary screw chillers located inside a lower level mechanical room and the resultant indoor noise. Case History 2 describes the pure tone noise problems associated with a 125 ton (440 kW), air-cooled rotary screw chiller located on a single-story roof and the resultant neighborhood outdoor noise.

Case History 1

This case history involves the water-cooled rotary screw chiller shown in *Figure 1*, which is one of two identical chillers located in a small, but separate, lower level mechanical room. The separate room provides compliance with ASHRAE Standard 15-1994, *Safety Code for Mechanical Refrigeration*,[1] and other jurisdictional codes. The mechanical room walls are constructed with concrete block, and one wall contains two sets of double wood doors for machinery access.

No ceiling was provided originally in the chiller room. However, the metal pan/concrete deck separating the chillers from the floor above is sprayed with 2 in. (51 mm) thick, acoustical-type fireproofing. The floor above the chiller room contains carpeted classrooms and secretarial offices separated by a corridor. Lightweight acoustical jacketing for the compressors, which had little acoustical value at the pure tone frequencies of interest, was provided from the factory at the time of chiller installation.

After operational startup and commissioning of the chillers, noise complaints were received from maintenance staff who visited the chiller room as part of their regular maintenance activities and

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from occupants in classrooms and offices on the floor above the chiller room. A noise survey revealed the following:

1. One-third octave-band sound pressure levels measured inside the chiller room indicated a very high pure tone noise at the 250 Hz, 1/3 octave band as shown in *Figure 2*. Levels varied in the room from 95 dB to 112 dB due to "standing waves"[2] at different locations, indicating numerous peaks and valleys at 250 Hz. Sound pressure levels at each location also varied between 2 dB and 16 dB over a 20-second interval indicating that the 250 Hz pure tone was time-modulated constantly by the slightly different rotor speeds. This is caused by the motor drive slip frequencies.

The frequency of the 250 Hz pure tone is consistent with the rotary screw configuration (4 lobes × 3570 rpm/60 = 238 Hz) and the second harmonic of the open drive motors (4 × line frequency of 60 Hz = 240 Hz).

The Air Conditioning and Refrigeration Institute test method (ARI Standard 575-95, *Method of Measuring Machinery Sound Within an Equipment Space*) for measuring chiller sound *averages* the 1/3 octave data over the entire chiller. This average sound data also is combined into octave bands for publishing purposes. Considering the averaging effects of this test method, combined with the increase in sound pressure levels due to the small size of the mechanical room and the hard reflective walls and floor, the *actual* field measurements resulted in noise that exceeded the manufacturers' published data by as much as 14 dB in one location. Because manufacturers' published data usually is averaged by ARI test methods at numerous locations around a chiller, and is tested outdoors or in a very large mechanical room, a single measurement point inside a small mechanical room never will duplicate the manufacturers' data.[3]

2. One-third octave-band sound pressure levels of 53 dB to 63 dB on the upper floor also indicated a pure tone noise at 250 Hz with level fluctuations of about 10 dB over the same 20-second interval measured in the chiller room. These sound pressure levels are overlaid with the chiller room noise in *Figure 2*. Sound pressure level measurements taken at various locations on the upper floor also showed "standing waves" with variations between the peaks and valleys of the standing waves.

Vibration measurements (taken in units of root-mean-square, "g"s of acceleration) at the 250 Hz, 1/3 octave band on the chiller housing, pipe hangers, and on the floor above the chiller room indicated that noise also was transmitted "structurally" through chiller piping support rods and hangers. This path of the structural-borne noise is from the chiller evaporator shell, through the chilled-water piping (both through the pipe wall and waterborne), clevis pipe hangers and support rods, and then re-radiated from the structural deck. Although this appears to be an unlikely path for the noise, the pipe-transmitted path actually is very common for chillers and pumps. The data also suggests that braided steel flexible connectors may not be able to reduce pure tone noise traveling through piping, and it points out the difficulties of maintaining proper hanger rod alignment on spring isolation hangers, which were used to support piping.

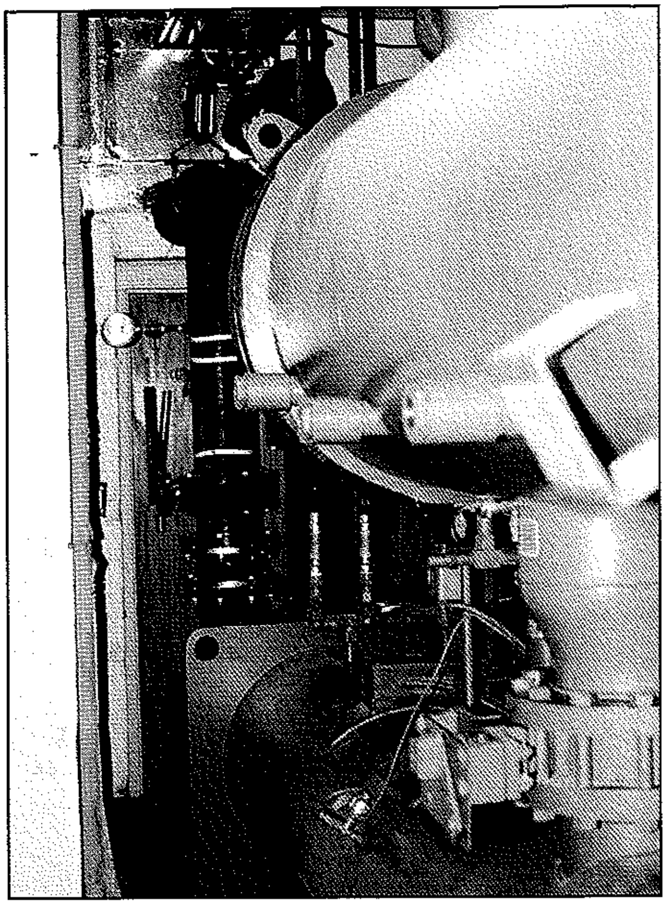

Figure 1: Indoor water-cooled rotary screw chiller.

The time and space variation of the 250 Hz pure tone made separation of the airborne noise from the structurally transmitted noise difficult to determine. Therefore, it was decided to reduce the airborne noise component first and then review the results before trying to control the structurally transmitted noise. To reduce reflected noise levels inside the chiller room for the maintenance staff, all available wall surfaces were covered with 2 in. (51 mm) thick rigid fiberglass over 0.75 in. (19 mm) deep furring strips. This reduced the 250 Hz pure tones inside the chiller room by approximately 6 dB. To reduce airborne noise through the concrete deck, a suspended drywall ceiling overlaid with 6 in. (152 mm) thick batt insulation was installed in the entire chiller room at 18 in. (457 mm) down from the concrete deck. The combination of wall treatment and suspended ceiling resulted in approximately 9 dB noise reduction at 250 Hz on the floor above the chiller room.

Reducing the structural-borne noise to the floor above the chiller room was somewhat more difficult. The multitude of spring isolation hangers used to support chilled water, condenser water and vent piping for two chillers and connected pumps resulted in numerous readjustments of the isolation hangers with only partial success.

As a last resort, single-sphere, neoprene flexible pipe connectors were installed in place of the braided steel connectors to reduce the structural-borne noise. The additional noise reduction (5 dB at 250 Hz) led to the conclusion that neo-

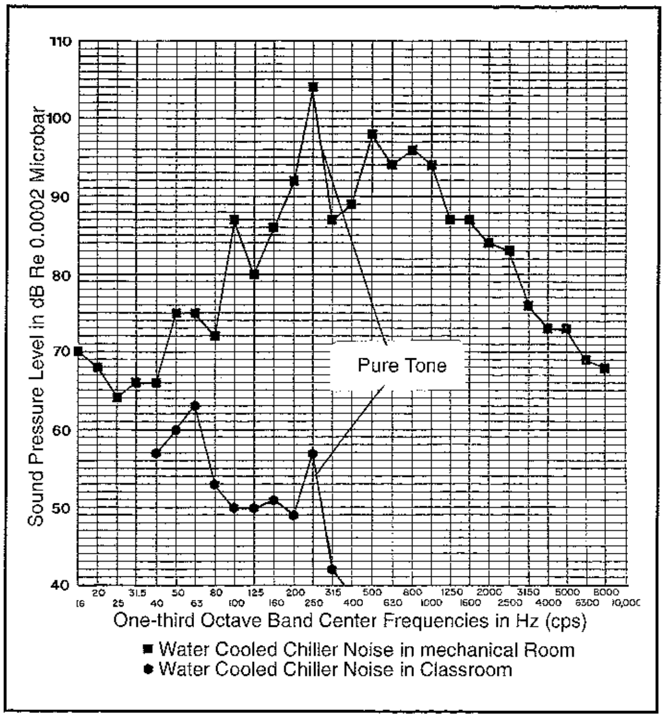

Figure 2: Water-cooled chiller noise in a mechanical room and in a classroom.

prene flexible pipe connectors are preferred over braided steel connectors for tonal noise transmitted along the pipe wall. However, the pure tone noise traveling through the direct water path is very difficult to reduce, and most likely is responsible for the limited amount of noise reduction gained through the neoprene flexible connectors. It should be noted that neoprene-type flexible pipe connectors have a high temperature limit dependant on line pressure. This author has experience in other installations that indicates the actual high temperature limit is 20°F to 30°F (7°C to 1°C) lower than the manufacturer's published data, which may make neoprene connectors unacceptable in some hot fluid applications.

Case History 2

Figure 3 shows the roof-mounted, air-cooled rotary screw chiller. The closest residential property line is approximately 300 ft (91 m) from the chiller. However, the chiller noise actually impacted the residential area up to 1/8 mile (0.2 km) away. The terrain between the chiller and residential area is mostly flat, with the chiller being elevated one story above grade. Two- and three-story buildings behind the chiller are reflective surfaces for the sound so that the pure tone noise also is "reflected" toward the residential area. A direct line of sight exists between the chiller and surrounding residences, with trees as the only barriers. The residential neighborhood is part of a larger farming community where the summer background sound level is very low, except for the occasional cricket or locust.

Soon after unit startup, noise complaints were received from several of the neighbors. One-third octave-band measurements taken at 3 ft (0.9 m) from the chiller compressors (*Figure 4*) revealed that the second harmonic of the rotor frequency (2 ×

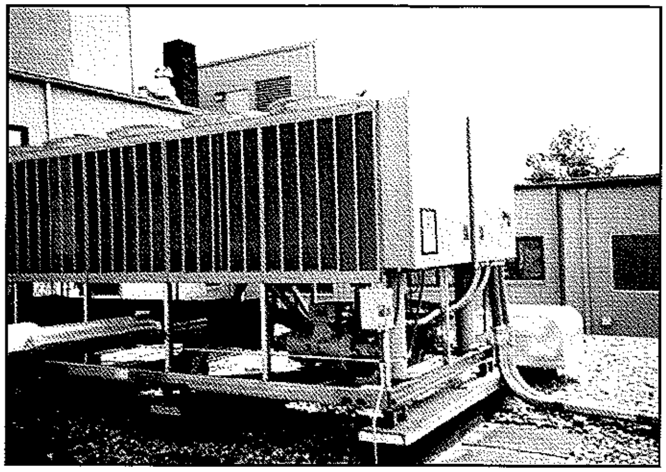

Figure 3: Outdoor air-cooled rotary screw chiller.

3600 rpm × 5 lobes/60 = 600 Hz) was the predominant pure tone noise showing as a peak at the 630 Hz, 1/3 octave. Octave-band sound pressure level measurements at 3 ft (0.9 m) from the chiller compressors (near field), and the primary complainants' property at 1/8 mile (0.2 km) (far field), showed a 33 dB difference in the sound pressure level at the 500 Hz octave band. The near field measurements were taken at 4 ft (1.2 m) above the one-story roof, and the far field measurements were taken at 4 ft (1.2 m) above ground level. There was a noticeable variation in the 500 Hz octave-band sound pressure level in the far field of approximately ±3 dB. One possible explanation for this variation and the relatively low noise attenuation with distance is that direct sound and reflected sound from the ground surface combined at the 1/8-mile location, continuously adding and canceling over time (commonly referred to as constructive or destructive interference).

To reduce the pure tone noise, the chiller screw compressors and connected oil separator were wrapped with acoustical jacketing. The jacketing consisted of a 0.5 in. (12.7 mm) thick polyester foam decoupler and 2 lb/ft^2 (9.76 kg/m^2) outer layer of vinyl loaded with barium sulfate. The jacketing was tailor-made in pieces to the specific chiller compressor and oil separator by a specialty manufacturer working in conjunction with the chiller manufacturer. Even though the pieces were made to fit the chiller components, gaps and cracks at seams were inevitable. The 1/3 octave-band measurements taken after the jacketing was installed showed a 6 dB noise reduction at the 630 Hz, 1/3 octave band. This data is superimposed with the "before" 1/3 octave data in *Figure 4*.

As a result of this work, most neighborhood complaints were reduced but not eliminated. The fact that the complaints were not eliminated likely is due to the very low background sound levels in the community. This makes the presence of a pure tone noise more prominent. Additional noise reduction can also be achieved with a barrier constructed around the chiller. An acoustical barrier[4] would have to extend several feet higher than the chiller and extend down to the roof, and be airtight in a horizontal direction, to be effective.

How to Evaluate Pure Tone Noise

During the design process, the HVAC engineer should properly select and size equipment and also evaluate equipment for prominent pure tone noise or other noise problems The only data that can assist in pure tone noise determination is a manufacturers' 1/3 octave-band or narrowband data, either in sound power or sound pressure levels Manufacturers typically measure equipment sound in 1/3 octaves, but this 1/3 octave-band data is currently difficult to obtain, and narrowband data rarely is available However, if the selected equipment will be located where it may have a noise impact on indoor occupants or outdoor neighbors, a mandatory requirement for submitting 1/3 octave data must be stated explicitly in the design documents.

Once 1/3 octave-band data is obtained from an equipment manufacturer, it must be analyzed to determine the presence of "prominent" pure tones, which can result in noise complaints. To accomplish this, the engineer should first divide the 1/3 octaves into three categories: 500 Hz to 10,000 Hz, 160 Hz to 400 Hz, and 25 Hz to 125 Hz. (Some manufacturers' data may not be available below 125 Hz due to the limited size of their indoor test facility.) If the 1/3 octave dB level of any band between 500 Hz and 10,000 Hz exceeds the "arithmetic average" of the two adjacent 1/3 octave bands on either side by 5 dB or more, then a prominent pure tone will exist. Likewise, exceeding 8 dB between 160 Hz and 400 Hz, and 15 dB between 25 Hz and 125 Hz, will show that a prominent pure tone exists.[5] When a prominent pure tone equals or exceeds the 5 dB, 8 dB or 15 dB criteria, there is a very high probability that a noise complaint will occur.

In the special case of selecting air- and water-cooled chillers, the chiller manufacturers' 1/3 octave data for screw chillers should be compared to the 1/3 octave data for equivalent reciprocating and centrifugal machines. If the chiller location can impact a noise sensitive area, and the screw chiller data indicate the presence of prominent pure tones, then consideration should be given to selecting alternative chiller types, chiller relocation, or to incorporating special noise controls If the screw chiller is already installed, then the methods described previously offer a means of anticipating noise complaints by taking field measurements in 1/3 octave bands at the receiver location and then by comparing the results to the 5 dB, 8 dB and 15 dB criteria

Using 1/3 octave-band data for design or measurement purposes adds the ability to convert the 1/3 octave data into full octave bands for comparison to RC criteria.[6] In the reverse calculation, 1/3 octave-band data cannot be calculated from octave bands Therefore, the determination of engineering controls for pure tone noise sources would be difficult with only full octaves as a database.

Conclusions

In noise critical installations, HVAC engineers should mandate the submittal of 1/3 octave-band sound data from manufacturers (currently ARI 370 for air-cooled chillers and ARI 575 for water-cooled chillers). This data must include all of the measurement points and should not be restricted to the average of the data points. The conditions of measurement also must be submitted so that the design engineer can correct the data for the specific application. If manufacturers' data indicates a prominent pure tone, the HVAC engineer must be prepared to provide the proper acoustical remedy. Alternate equipment types and proper equipment location should be contemplated early in the design process. For indoor equipment, both airborne and structural-borne noise controls must be considered, as demonstrated in Case History 1 For outdoor equipment, the HVAC engineer should try to use natural barriers such as buildings or berms, or add equipment barrier screen walls Other possible controls at the source of noise, such as the acoustical jacketing demonstrated in Case History 2, are often available from equipment manufacturers.

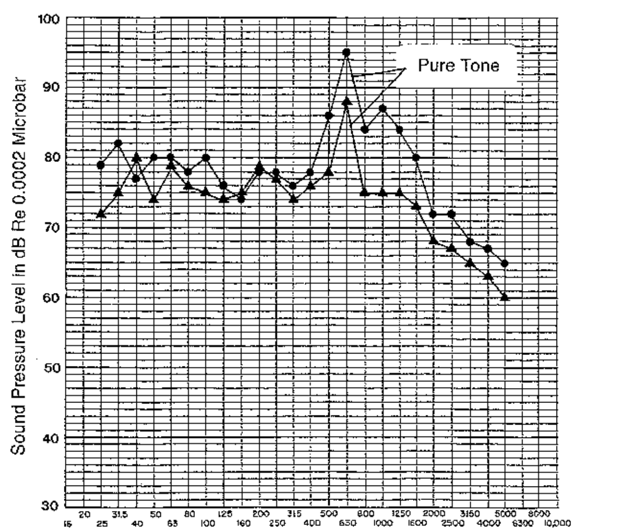

Figure 4: Air-cooled chiller noise measured at compressors.

References

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2. Beranek, L. 1987 *Noise and Vibration Control*

3. Ebbing, C. and W. Blazier 1998 *Application of Manufacturers' Sound Data*. ASHRAE.

4. *1995 ASHRAE Handbook—Applications*, Chapter 43, p. 43.28–29

5. 1977 *Illinois Pollution Control Board Rules and Regulations*, Chapter 8, Noise Regulations.

6. *1995 ASHRAE Handbook—Applications*, Chapter 43, p. 43.3–4