Noise Control of Air-Cooled Heat Exchangers

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Paper for a Presentation at a Session on Noise — Results of Various Approaches to Control

Presented at the

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ABSTRACT

This paper deals with noise caused by air-cooled heat exchangers. Points discussed are the source and mechanism involved, as well as the relationship of the variables affecting such noise. Air-cooler directionality, specification objectives, available range of noise and costs of quieting are discussed.

INTRODUCTION

Air-cooled heat exchangers (FIN-FAN® units) are being used more extensively year by year in refineries, petrochemical plants and many other places where heat must be removed from a fluid. At the same time, several factors are coinciding to focus attention on noise generated by these exchangers. Normally speaking, FIN-FAN® units are less noisy than several other commonly occurring noise sources in plants (vents, furnaces, gear pumps, etc.). However, due to the dependency of air coolers on unrestricted air flow, it is more expensive to achieve a given noise reduction on FIN-FAN® unit than on most other types of equipment. In many cases, one can simply contain the noise. In the case of furnaces, the limited air flow can be conveyed through a duct, which gives the required attenuation.

The literature reflects the variety of opinions presently held as to the cause of FIN-FAN® unit's noise. Several factors that have been considered are:

1) Rotating forces (blade pass frequency plus harmonics)

2) Random vortex shedding

3) Air turbulence

4) Air flow over fins

5) Bearing rumble

6) Resonant effect of air-cooler plenum chamber

There have also been several equations developed which attempt to predict the noise resulting from FIN-FAN® units with large axial flow fans.

Beranek (1) gives:

$$\text{PWL (overall)} = 138 + 20 \log \text{hp} - 10 \log q \qquad (1)$$

Where:

PWL = sound power level (re 10^{-12} watts), dB
hp = motor shaft horsepower
q = acfm, fan discharge
(Intended for both axial flow and centrifugal fans)

Allen (2) gives:

$$\text{PWL (overall)} = 68 + 10 \log q + 20 \log p \qquad (2)$$

Where:

p = static pressure, inches H_20

Gordon, et al (3) gives:

$$\text{PWL (125 Hertz)} = 60 \log (Vt \times 10^{-3}) + 10 \log \text{acfm} - 7 \qquad (3)$$

Where:

Vt = tip speed, ft/min

Underwood (4) gives:

$$\text{PWL (125 Hertz)} = 55 \log_{10} (Vt \times 10^{-3}) + 20 \log \text{hp (abs)} \qquad (4)$$

$$-10 \log \text{acfm} + 10 \log (N) - 96$$

Where:

N = number of blades

Seebold (5) gives:

An equation to predict the SPL at a specific point with respect to the unit. For induced-draft units, at three feet below the tubes:

$$SPL(overall) = 63 + 20\,log\left[hp \times \left(\frac{Vt}{10000}\right)^{4/7}\left(\frac{6}{N}\right)\left(\frac{12}{D}\right)^{4/7}\right] \qquad (5)$$

Where:

SPL = sound pressure level (re 0.0002 microbars), dB
hp = hp, consumed power basis
D = fan diameter, ft.

For forced-draft units, Seebold adds about 5 dB.

Sharland (6) gives:

$$\text{PWL (overall)} = 56 \log Vt + 10 \log [Cm^{0.6}\, L\, N] + A_2 \qquad (6)$$

Where:

Cm = blade mean chord width, ft.
L = blade length, hub to tip, ft.
N = number of blades
A_2 = constant

Another correlation by Lawrence (7) gives:

$$\text{PWL (overall)} = 95 + 10 \log hp \qquad (7)$$

Considering these equations in order, equation (1) states that the noise energy is proportional to the horsepower raised to a power of 2.0 and to acfm to a power of -1.0 and is unaffected by tip speed, ΔP, fan size, number of blades, etc.

Equation (2) states that the noise energy is proportional to acfm to a power of 1.0 and ΔP to a power of 2.0 and ignores all other factors.

Equation (3) states that the noise energy is proportional to tip speed to a power of 6.0 and acfm to a power of 1.0 and ignores all other factors.

Equation (4) states that the noise energy is proportional to tip speed to a power of 5.5, hp to a power of 2.0, acfm to a power of -1.0, number of blades to a power of 1.0 and ignores other factors.

Equation (5) states that the noise energy is proportional to hp to a power of 2.0, tip speed to a power of 1.143, fan diameter to a power of -1.143, and number of blades to a power of -2.0.

Equation (6) states that the noise energy is proportional to tip speed to a power of 5.6, mean blade chord width to a power of 0.6, and blade length and number of blades to a power of 1.0.

Contributing to this diversity of opinion, data taken by one group may be evaluated with considerably different results by another group. This problem arises because an air cooler is a directional noise source, and does not lend itself readily to techniques, the aim of which is to assign a PWL value to one or two points within its bounds.

The method behind most of these correlations is simply to develop an equation which is reasonable and most closely matches test results. The work of Sharland reflects a wider range of variables due to being based on a small scale setup in which the variables could be changed, one at a time.

It is suggested that the method of correlating data on existing units is self-limiting, in a sense. That is, no points are represented beyond present practice, whether tip speed, horsepower, acfm, etc. No data has been made available in the literature in which one or more representative units were tested, changing one variable at a time.

Only after such a program is carried out will there be reasonable agreement on the relative influence of the several variables involved. In this respect, the work of Sharland is important (even though his experiments were on a rotor of 6" diameter) because this rule was followed.

Following are several observations reflecting the writer's opinions as to the source of FIN-FAN® unit’s noise and the relationship of the several variables.

1) FIN-FAN® unit’s noise is fan-generated noise. Less than one percent of the noise energy can be related to panel or structural vibration. However, structural components may influence fan-generated noise by creating additional turbulence around the blades.

2) This fan-generated noise is caused by vortex shedding and air turbulence. It is broad band without discrete peaks (other than the blade pass frequency, which is normally below 31 hertz).

3) The evidence is that this noise is proportional to tip speed to a power of 5.6 to 5.8.

4) The parameter of next importance is neither horsepower nor acfm, but is pressure differential across the fan. The exponent on ΔP is about 1.4 but varies inversely with tip speed.

5) Acfm is of very little importance, as is horsepower. The acfm can be varied over a 10/1 range with very little effect on noise, assuming other variables are held constant, and the stall range is not reached.

6) Regarding fan size, inadequate data exists to do more than make a guess at this effect.

7) Regarding number of blades, the evidence is this is not significant if one avoids the stall range. This point will be expanded later in the paper.

Regarding the previous seven statements pertaining to FIN-FAN® unit's noise, some discussion is in order. The fourth statement is that fan noise is proportional to the pressure differential across the fan to an exponent of 1.4, and that this exponent varies inversely with tip speed. This may readily be demonstrated by operating a given fan at four states as follows:

a) Vt_l, ΔP_1

b) Vt_l, ΔP_2

c) Vt_2, ΔP_1

d) Vt_2, ΔP_2

The pertinent noise data of these four tests will show a greater variation of noise caused by the change in ΔP at the lower tip speed than at the higher tip speed.

The fifth statement is that both acfm and horsepower are of little importance. While this appears to be contrary to the mass of field data accumulated by others, it is suggested that this field data reflects a narrow range of fan performance as to pressure differential, fan horsepower, and blade pitch angles. While it may be argued that this narrow range covers 95% of the useful range as far as FIN-FAN® units are concerned, any misleading assumptions will delay progress toward a more precise understanding of fan noise. It is suggested that this question may be fairly easily resolved by varying the air quantity on a fan under test while holding other factors essentially constant. A variation of horsepower while holding other factors essentially constant poses a problem. Therefore, evaluation of horsepower effects may require cross-checking two setups in which horsepower and other variables were allowed to change.

The seventh statement is that the number of blades in a fan is not significant if one avoids the stall range. On this point, the literature reflects opinions in both directions. Some contend that an increase in the number of blades increases noise, while others hold that noise is decreased. Again, this point may be resolved by tests in which the only variable is the number of blades.

It is pertinent to inject, at this point, that one must not equate the blades of different design. That is, a 14 ft. diameter fan with 4 blades from manufacturer A may be more efficient and have a higher pressure capability at a given tip speed than a comparable six-blade fan from manufacturer B.

Regarding the noise field around a FIN-FAN® unit, several observations are in order. Since the noise generated is proportional to blade velocity to a power of 5.8, a point at the tip of a blade is generating 1.8 times as much noise as a point 9/10 out from the center toward the tip. Thus, approximately 80% of fan noise is generated in the outermost 20% of the fan diameter. On a 14 foot diameter fan, this zone would be defined as being 11.2 ft. I.D. and 14 ft. O.D., or the outer 1.4 ft. of each blade.

Several factors are important to an understanding of the directionality of FIN-FAN® units. First, the previous paragraph discussed the importance of the outer segments of the blades as the origin of most of the noise. Each blade, therefore, may be considered as having a point noise source located at a point 9/10 from the hub to the tip and rotating with the blade. Second, the tube bundle causes no significant attenuation of the noise passing through it. One must remember that the tube bundle is relatively porous, having approximately 50% free area at the most restrictive points. Third, the sheet metal containing the air flow between the fan and bundle also contains the noise so that most of the noise must escape through the air inlet or exit areas. Therefore, only if the noise is measured very close to the containing panels (less than three feet) will the noise transmitted through the panels exceed the noise from the fan which passes through the air inlet or exit. Exceptions to this rule will occur if the panel material used is too thin. There may be panel rattle or "popping" of panel areas due to passing from a convex to a concave state, if light material is used. The author's company presently uses 8 gauge (.164) material or heavier to eliminate this possibility.

Tests show that a point 25 feet horizontally from a fan will indicate 5-8 dB less than a point 25 feet above or below a fan, assuming one layer of sheet metal of normal construction serves to baffle the horizontal case. Therefore, any specification which covers a zone not shielded from the fan by continuous sheet metal must take these factors into account. Many specifications are written to require testing for maximum SPL at three feet from any bounding surface. This translates into three feet below the fan guard on forced-draft units, and three feet below the tubes on induced-draft units. (The area above the fan on induced-draft units is excluded from the specification.)

Thus, on induced-draft units, the measurement point is 8-9 feet (average) from each rotating noise source, and on forced-draft units, 6-7 feet. Therefore, the latter case will give readings about 3 dB higher than the former.

In order to better visualize the pattern of noise surrounding a FIN-FAN® unit, consider Figure 1. This is a scale drawing of an induced-draft unit. It is assumed that each unit (2 fans) is generating a sound power level of 100 dB (97 dB per fan). This power level would be based on one of several hemispherical evaluation procedures. The frequency could be any one of the first five octave bands.

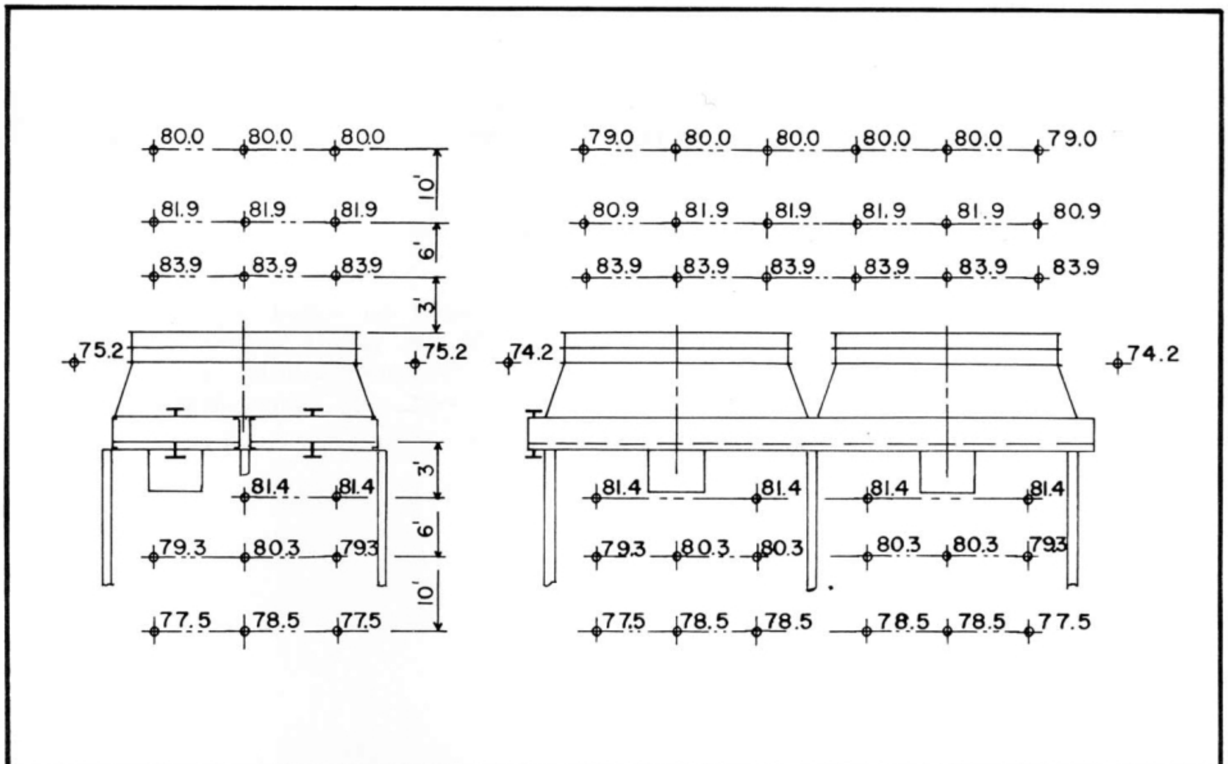

FIGURE 1—SPL READINGS NEAR AN INDUCED DRAFT AIR COOLER OF 100 dB PWL (97 dB PWL PER FAN).

The horizontal plane three feet below the tubes yields maximum SPL readings of 81.4 dB, while a point three feet above the fan ring gives 83.9 dB. Also, notice that a point three feet out from the transition gives 75.2 dB, even though it is much closer to the noise source, but is shielded from the line of sight transmission. Figure 2 shows similar information for a forced-draft unit.

Equipment noise specifications usually are intended to limit the noise either immediately adjacent to the equipment or in a zone such as a neighborhood some distance away. The control of noise near the equipment may be motivated by one of several federal and state laws such as the Walsh-Healey Act. A limit of 90 dB(A) on walkways beneath FIN-FAN® units calls for little or no decrease of noise with respect to induced-draft units of standard design and only 3-4 dB decrease on forced-draft units.

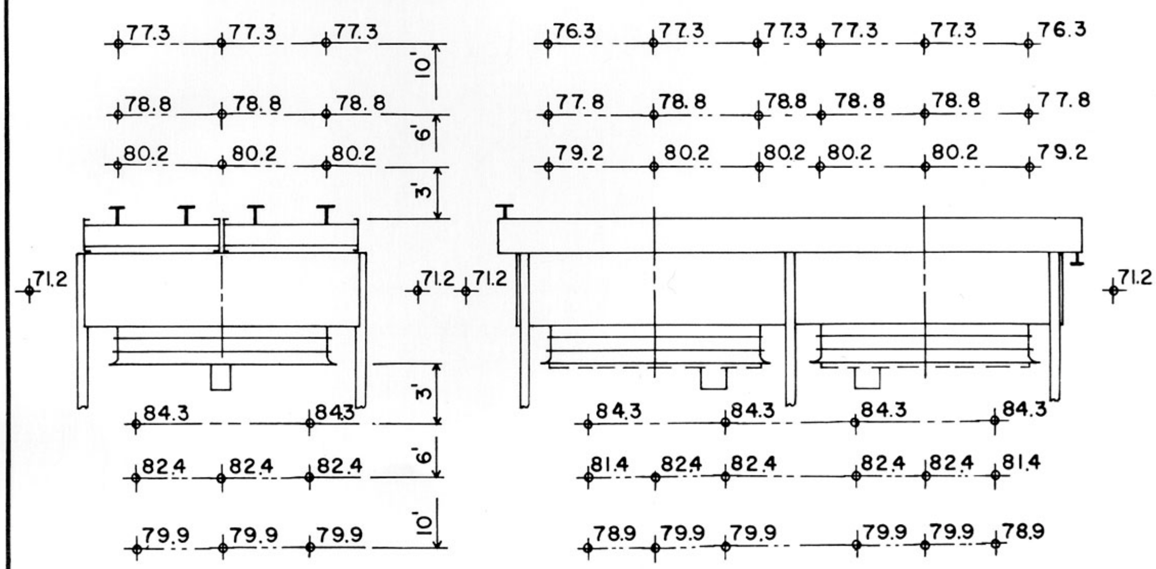

FIGURE 2 - SPL READINGS NEAR A FORCED DRAFT AIR COOLER OF 100 dB PWL (97 dB PWL PER FAN).

However, controlling neighborhood noise in areas adjacent to the plant boundary may impose much more extreme limitations on equipment design. For instance, a refinery in England is located adjacent to a housing project in which the noise level is approximately 70 dB(A). It would be necessary to reduce the refinery noise to 1% of its present level to achieve 50 dB(A) in the housing project, based on the assumption that the refinery is the primary cause of the present noise levels. It is very doubtful that the expense of such a change can be justified. It is more likely that the refinery would be salvaged and relocated, or continue to operate on a reduced basis.

Several refineries now in either the planning or construction stage have total sound power level limits which are designed to insure maximum sound levels in adjacent communities of 35 dB(A) (at night), 40 dB(A), and 50 dB(A). If a law exists governing noise in a residential area, it is not likely to allow noise in excess of 50 dB(A). A common practice is to allot 50% of the allowable noise to FIN-FAN® units when the noise specification requires that the total plant noise PWL be 110-120 dB (A).

If it is assumed that a plant has an overall PWL limit of 115 dB (A) and a limit of 112 dB(A) is applied to 50 FIN-FAN® units, then the limit per FIN-FAN® unit is 95 dB(A). This means that each FIN-FAN® unit must cause 2.5% as much noise as one of standard design. (Standard design is approximately 3 dB(A) PWL per FIN-FAN® unit.) This decrease of 16 dB will increase both the capital cost and plot plan of the FIN-FAN® unit for a given duty. The increase should fall in the range of 20-30%.

Recent progress in fan development promises a reduction in the added cost to achieve a given noise decrease, as well as reducing the required increase in plot plan. In fact, it is now possible to achieve 8-10 dB reduction below standard design noise levels with no increase in plot plan.

Both in the United States and in Europe, legal limits are being imposed at the boundaries of installations that require 5-10 dB less noise at night than in the daytime. Variable pitch fans give some reduction of noise (as well as saving power) as the ambient temperature decreases at night. For instance, based on the assumption that the MTD of a group of FIN-FAN® units is 50ºF or less, an ambient temperature decrease of 15ºF would permit a fan power reduction of 40-45%.

SUMMARY

The control of FIN-FAN® unit's noise (and, indeed, practically all noise) is primarily an exercise in economics, since it is possible to achieve any desired noise level if the cost of the noise reduction is justifiable. This becomes obvious if one considers that somewhere between minimum cost equipment (having a relatively high air flow) and maximum equipment size and cost (no fans), one must span the entire range of possible FIN-FAN® unit's noise down to 0 dB. As a partial illustration of this point, the author's Company has built and demonstrated FIN-FAN® units having reasonable air flow levels which generated 1/1000 as much noise as a standard unit. This is a reduction of 30 dB.

Further developments in this field will probably bring forth a new generation of fans, these fan designs being aimed at not only moving air efficiently but quietly. Therefore, the better we understand axial flow fan noise, the more effective our efforts at reducing it will be.

REFERENCES

1. Beranek, L. L., Noise Reduction, 1960, page 544.

2. Allen, C. H., "Noise From Air Conditioning Fans", Noise Control, Vol. 3, pages 28-34, January, 1957.

3. Gordon, C. G., and Randall, M. D., "The Noise of Air Fin Coolers", I.Mech.E. Symposium, September 24, 1970.

4. Underwood, P. C. V., "Working to Noise Specifications - A Contractor's Approach", I. Mech. E. Symposium, April 27, 1971.

5. Seebold, James G., "Process Plant Noise Control at the Design Engineering Stage", ASME Paper No. 70-Pet-11.

6. Sharland, I. J., "Sources of Noise in Axial Flow Fans", J. Sound and Vibration (1964), I (3), 302-322.

7. Lawrence, H. R., "Near Field Sound Pressure Measurements Around an Air Cooled Heat Exchanger", American Industrial Hygiene Conference, May 14,1969.