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THE COST OF NOISE

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ABSTRACT

Today, owner/operators, OEM's and suppliers are facing lower and lower near and far field noise limits with respect to their equipment. However, lost in this race to see who can out quiet who is the impact of cost. Specifically – the cost of noise with respect not only to fans, but the fan mechanical/structural parts as well

This paper will look at two specific applications, one a bank of induced draft air-cooled heat exchangers and the other being a set of field erected cooling tower cells. In both case studies, the cost of lower and lower near and far field noise will be evaluated with respect to the fan and mechanical and structural components.

INTRODUCTION

We will look at two case studies here, the first being a 11 bay bank of Air-Cooled Heat Exchangers (ACHEs) and the second being a 8 cell counter flow Cooling Tower (CT). Before getting into the descriptions of the equipment, we will review the assumptions that went into this analysis. The following assumptions apply to both systems unless otherwise noted:

- The total airflow and static pressure delivered by the fans is maintained as noise is reduced.
- Noise reduction is achieved solely by speed reduction of the fans and modification of the fans to different blade counts and blade types. No other noise abatement devices were considered and standard motors, gears or drives were employed.
- Near and far field noise predictions are for fans only. No attempt was made to assess noise generated by the drive/gear systems, motors or waterfall (cooling tower only).
- Inlet conditions and tip clearances of the fans remained constant.
- ACHE near field noise were predicted one meter below the ACHE bank center with all the fans running
- CT near field noise were predicted between cells 4 and 5 along the centerline of the towers, two meters above the deck level, with all the fans running.
- Far field noise for both the ACHE and CT were predicted at 100 meters perpendicular to the long side of the units, 2 meters above the ground.
- The noise correlations used were the same for both the ACHE and CT systems.
- No additional bays or cells have been added – plot area remains constant.
- Heat transfer surface remained constant.
- CT water flow remained constant.
- Tolerance on near and far field Sound Pressure Level (SPL) and Sound Power Level (PWL) noise predictions are +/- 2 dB(A).
- Tolerance on cost estimates is +/-10% to 15%.

Additionally the authors wish to emphasize that, with respect to cooling towers, only fan noise spectrum and reduction was considered here and this paper does not attempt to address the higher frequency water noise. While fan noise can be analyzed without changing the overall system resistance, water noise reduction/suppression requires adding attenuators to the inlet and/or outlets of the cooling tower. These attenuators increase the overall system resistance (i.e. increases motor power draw) and thus add a level of complexity that, while important, was outside the scope and purpose of this paper. However, the authors do agree that the subject of cooling tower water noise should be addressed in the future as a separate paper. Finally, it is acknowledged that other design options are available for noise reduction such as ACHE/CT redesign or other low noise technologies. But as with water noise, our scope here is solely limited to fan noise reduction.

AIR-COOLED HEAT EXCHANGER DESCRIPTION

The base design air-cooled heat exchanger (ACHE) described in this paper (Figures 1 and 2) is a grade mounted, carbon steel induced draft item built to the API-661 Standard (Reference 1). The ACHE has a thermal duty of 29.3MW (100 Million Btu/hr) cooling light gasoline from 60.3C (141F) to 37.8C (100F) at an ambient design temperature of 32.2C (90F). The item consists of 11 individual bays with 12.2 m (40.0ft) long 25.4 mm (1.0 in) OD carbon steel tubes with extended surface. The extended surface consists of extruded aluminum fins 15.9 mm (0.625 in) high fins spaced at 10 fins per inch. The tubes are spaced in an equilateral tube pitch of 63.5 mm (2.5 in). The individual bays are 4.98 m (16.34 ft) wide with an overall item with of 54.8 m (179.8 ft). Height from the bottom of the tube bundle frame to grade is 2.74 m (9.0 ft).

The base mechanical fan drive systems consists of 25 HP, 60 HZ, single speed motors, synchronous belt speed reduction, and 3.96 m (13 ft) fiberglass reinforced plastic (FRP) fans with 4 blades. Each bay has two mechanical drive systems with the entire item containing twenty-two such systems.

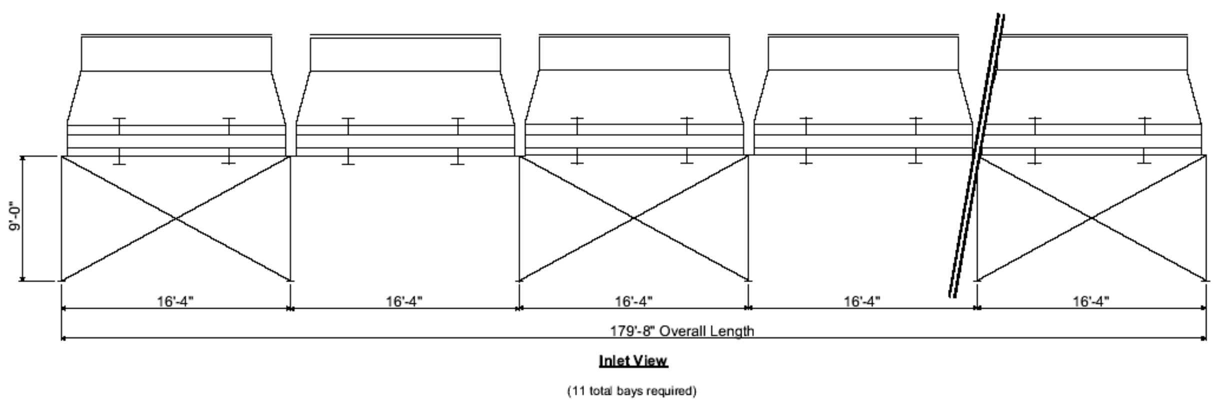

Figure 1. Induced air-cooled heat exchanger front (inlet) view

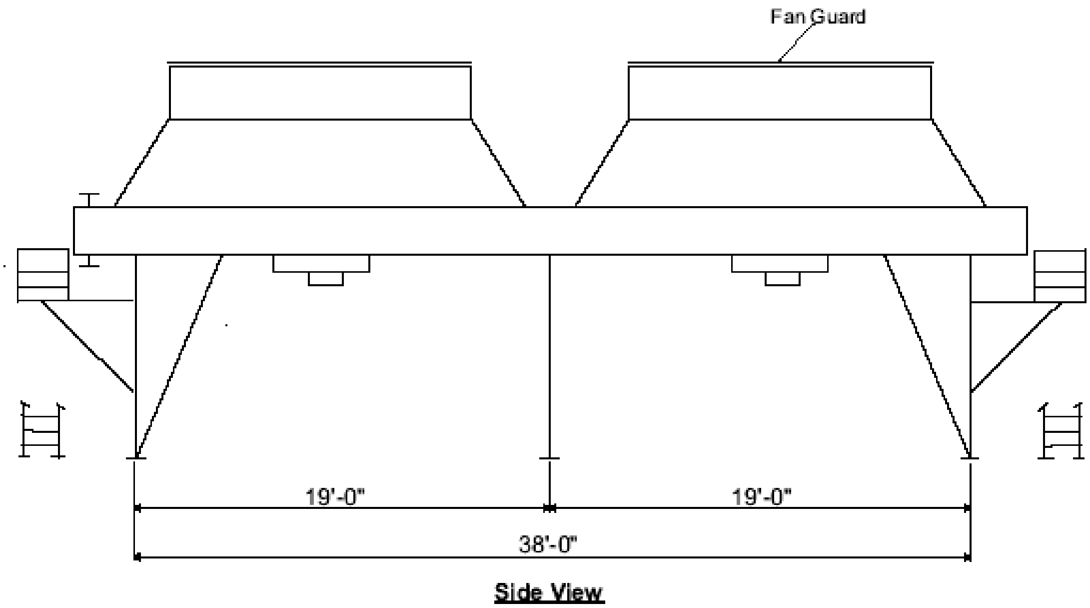

Figure 2. Induced air-cooled heat exchanger side view

COOLING TOWER DESCRIPTION

The base CT described in this paper is an 8 cell, in line, induced, mechanical draft cooling tower (Figures 3, 4 and 5) designed to the applicable CTI standards and guidelines (References 2,3). The tower structure is constructed with fire retardant FRP structural components and incorporates 304 SS hardware for all structural and mechanical connections. The roof deck is FRP with a non-skid surface applied. The interior and exterior casing is 12 oz – fire retardant (FR), FRP casing. The tower includes two – FR, FRP stairways, one at each end of the tower and one FR-FRP ladder and cage, located in the center of the tower. Cell size is 14.63 m x 14.63 m (48 ft x 48 ft), with 1.83 m (6 ft) of low fouling PVC film fill that is bottom supported. The drift eliminators are PVC, cellular type, 0.40 mm (0.015 in) thick; with a maximum allowable drift rate of 0.0015 % of design flow. The tower design flow is 401,254 L/min (106,000 GPM) total with entering hot water of 37.8 C (100 F), exiting cold water of 29.4 C (85 F) and a design wet bulb temperature of 25.6 C (78 F).

The base mechanical fan drive systems consists of 250 HP, 480 VAC, 60 HZ, single speed motors, double reduction gear reducers, composite drive shafts and 10 m (32.8 ft), FRP fans with 9 blades. The FR-FRP fan stacks are velocity recover type design, 10 m (32.8 ft) in diameter and 3.05 m (10 ft) in height. Controls consist of low oil level cut off switches and vibration switches with manual and remote reset features mounted on the gear reducer. The tower is installed on a customer supplied concrete basin with a basin depth of 1.22 m (4.0 ft).

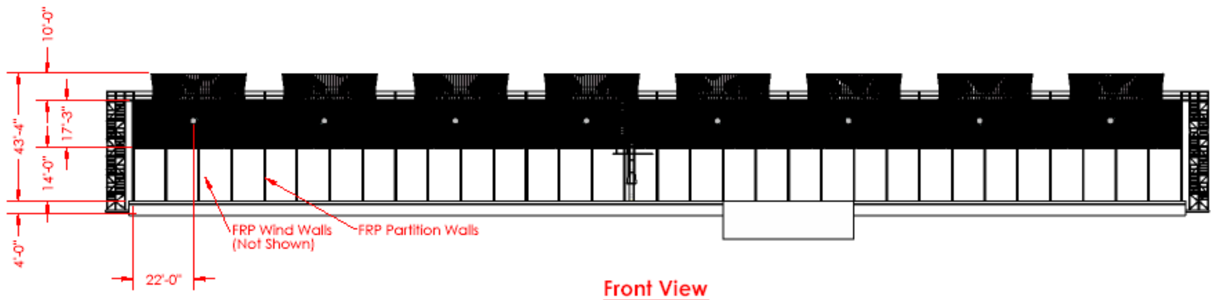

Figure 3. Cooling tower front view

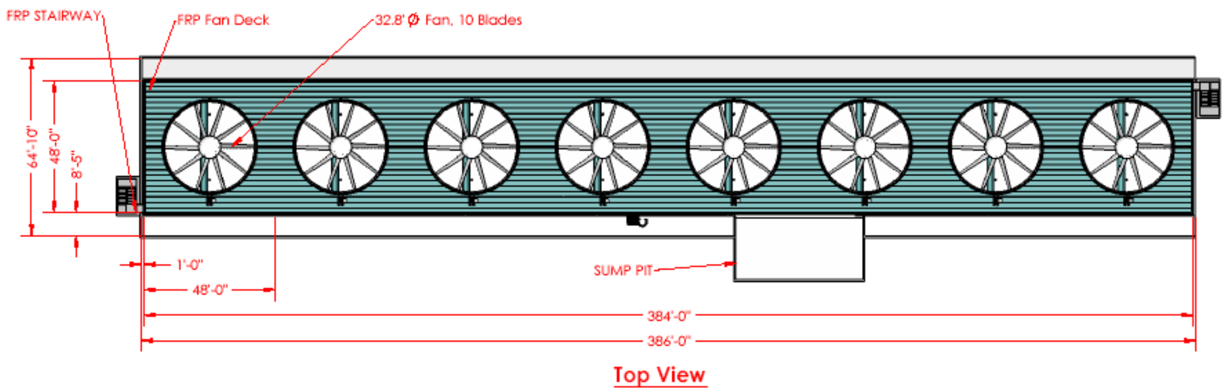

Figure 4. Cooling tower top view

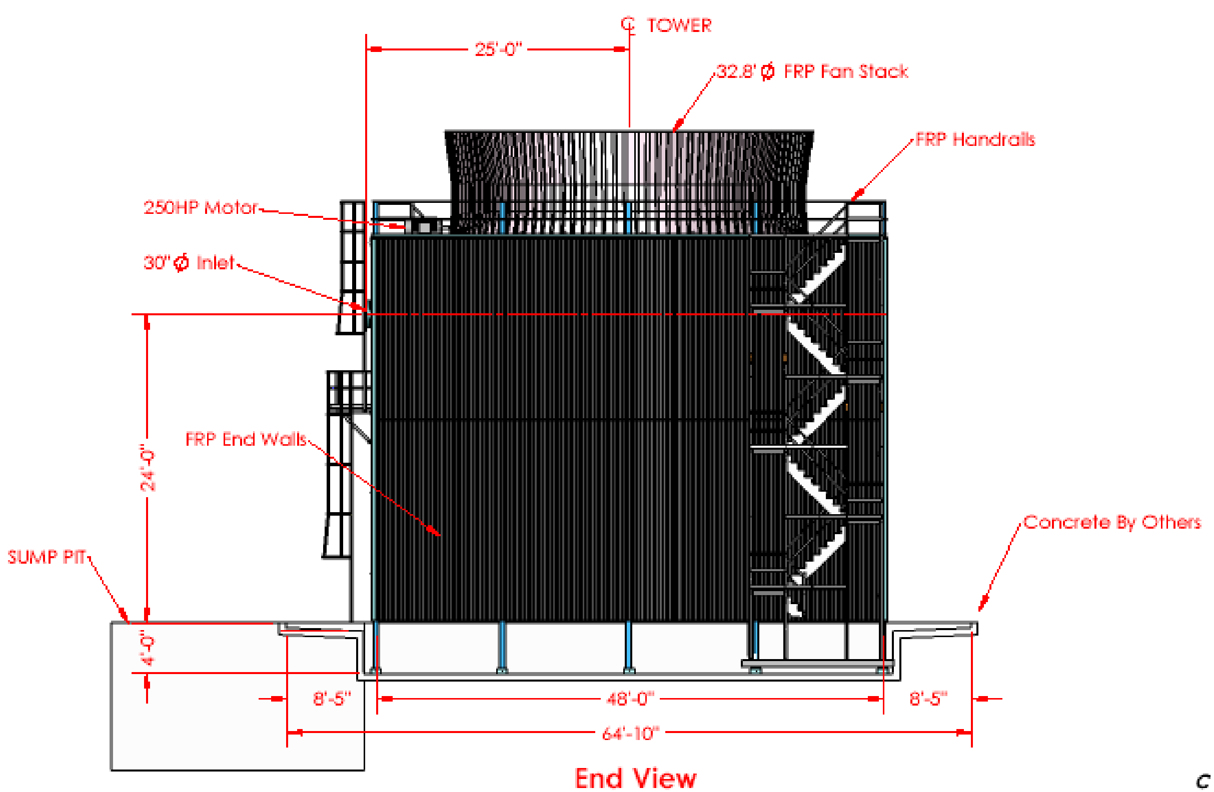

Figure 5. Cooling tower end view

CASE 1 – AIR-COOLED HEAT EXCHANGER

Table 1 lists the common operating parameters of all the ACHE fans studied. Table 2 lists the base case fan condition along with the lower noise options listed in order of decreasing noise.

Fan Parameter	Value or Description
Air Flow; m^3/sec (CFM)	72.53 (153,680)
Static Pressure; Pa (in H20)	129.5 (0.52)
Air Density; kg/m^3 (lb/ft^3)	1.157 (0.0722)
Inlet Bell Type	Rounded (R/D = 0.05)
Fan Diameter; m (ft)	3.96 (13.0)

Table 1. ACHE fan-operating parameters.

Fan Case	Fan Type	Fan RPM	Tip Speed m/sec (ft/min)	Shaft Power/Fan KW (HP)	Sound Power Level per Fan dB(A)
ACHE – Base	STD	296.9	61.6 (12,124)	13.7 (18.3)	102.1
ACHE – 1	STD	265.6	55.1 (10,847)	14.0 (18.7)	99.3
ACHE – 2	VLN	234.4	48.6 (9,573)	15.7 (21.0)	95.7
ACHE – 3	VLN	208.8	43.3 (8,527)	15.7 (21.0)	92.7
ACHE – 4	VLN	168.2	34.9 (6,869)	15.4 (20.7)	89.6
ACHE – 5	ULN	208.8	43.3 (8,527)	13.7 (18.3)	85.6
ACHE – 6	ULN	186.9	38.8 (7,633)	13.3 (17.9)	82.6
ACHE – 7	ULN	151.4	33.8 (6,659)	13.5 (18.1)	80.4
ACHE – 8	ULN	136.5	30.5 (6,004)	13.9 (18.6)	79.3
STD – Standard Noise, VLN – Very Low Noise, ULN – Ultra Low Noise					

Table 2. ACHE fan-operating conditions in order of decreasing noise.

As one can see from Table 2, the fan speed was lowered to approximately 50% of the base design while the fan shaft power remained fairly constant. Fan speed was varied in approximately 10% increments by a combination of drive ratios and motor speeds to achieve the required noise reductions. Multiple fan selections were performed for the STD, VLN and ULN fans types as applicable to achieve the overall most cost effective solution at that particular speed. Cost increases were estimated for fan Cases 1 through 8 taking into account changes (if any) in fan, motor, drives, mechanical structure and fan rings. The cost increase estimates were then plotted against the near and far field SPL noise predictions as shown in Figure 6 below.

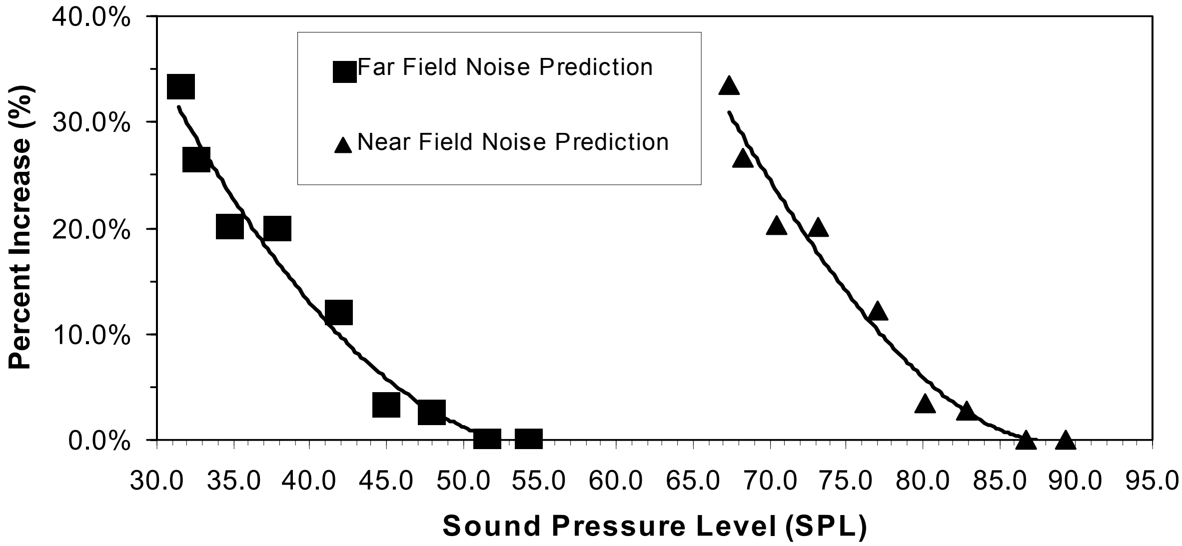

Figure 6. ACHE bay cost increase vs. near and far field SPL

The overall ACHE cost per bay significantly increases as noise is decreased with the lowest noise option increasing the per bay cost by approximately 35%. Given the non-linearity of the data (a general 3rd order polynomial was used to trendline the data) one can see that the cost increase accelerates faster for a relative steady decrease in noise. Put another way, the first 10 dB(A) of far field noise reduction increased the overall cost per ACHE bay by about 4% from the base design. However the next 10 dB(A) increased the cost 20% from the base design.

The majority of this increase is related to the fan and depending on the type of fan (VLN or ULN – See Figure 7 for a general size comparison) will range from 70 to 90% of the total increase. This is shown below in Figure 8 where the costs increases are plotted as a function of fan sound power level.

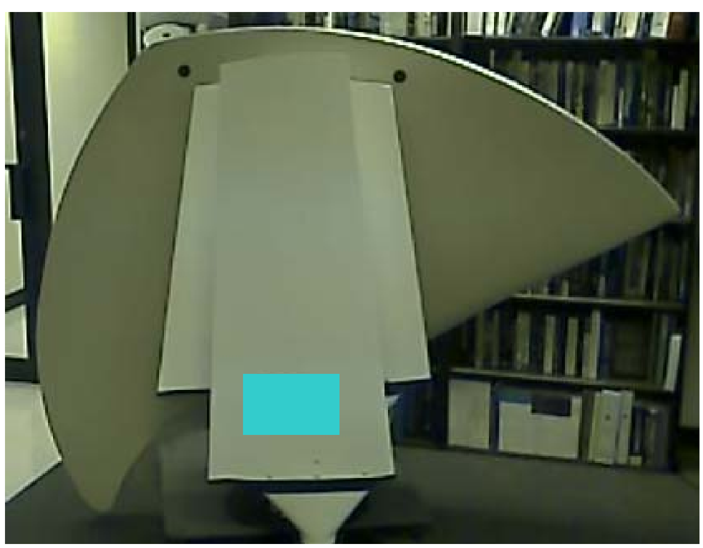

Figure 7. Size comparison of standard noise (foreground), very low noise (middle) and ultra low noise (back) fan blades

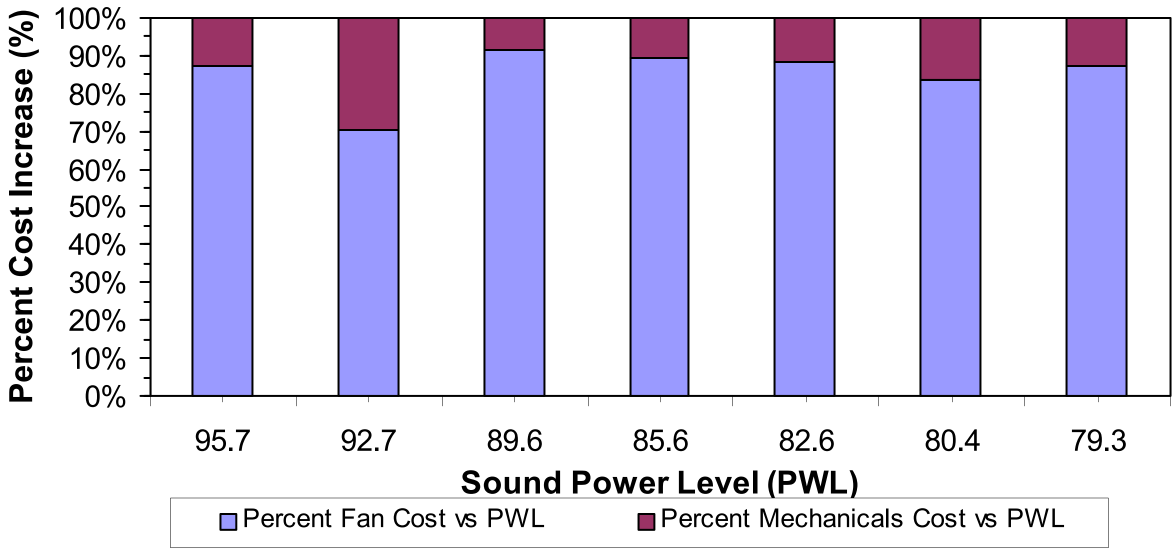

Figure 8. Percent cost increase split between fans and mechanicals

The variance shown in Figure 8 is simply a function of the design points chosen. For example, at the 92.7 dB(A) SPL point, nearly 30% of the cost increase is due to non-fan components due to the fact that the fan, in this instance, did not change as the speed was reduced. However, when looked at in total, this combination of fan, motor, drives and so forth provided the least amount of overall cost increase. Another factor is the fan material. Aluminum fans tend to be less costly than FRP fans and will move the cost split more towards the non-fan components. A good rule of thumb is that the fan will account for approximately 70% (Aluminum) to 85% (FRP) of the total cost increase as noise is reduced. However, every situation is unique and the owner/operator is encouraged to investigate all possibilities with the OEM/Supplier.

CASE 2 – COOLING TOWER

Table 3 lists the common operating parameters of all the cooling tower (CT) fans studied. Table 4 lists the base case fan condition along with the lower noise options listed in order of decreasing noise.

Fan Parameter	Value or Description
Air Flow; m3/sec (CFM)	556.3 (1,178,680)
Static Pressure; Pa (in H20)	157.4 (0.632)
Air Density; kg/m^3 (lb/ft^3)	1.123 (0.0701)
Inlet Bell Type	Elliptical (R/D = 0.10 x 0.15)
Fan Diameter; m (ft)	10.0 (32.81)

Table 3. Cooling Tower fan-operating parameters.

Fan Case	Fan Type	Fan RPM	Tip Speed m/sec (ft/min)	Shaft Power/Fan KW (HP)	Sound Power Level per Fan dB(A)
CT – Base	STD	112.9	59.1 (11,637)	119.4 (160.1)	106.0
CT – 1	STD	100.9	52.8 (10,400)	119.4 (160.1)	103.9
CT – 2	LN	90.2	47.2 (9,297)	121.8 (163.3)	101.5
CT – 3	VLN	79.5	41.6 (8,194)	123.9 (166.2)	97.3
CT – 4	VLN	72.4	37.9 (7,463)	128.0 (171.7)	96.0
CT – 5	VLN	70.0	36.7 (7,215)	135.4 (181.6)	95.4
CT – 6	ULN	72.4	37.9 (7,463)	175.7 (235.6)	93.5
CT – 7	ULN	70.0	36.7 (7,215)	198.3 (265.9)	92.6
STD – Standard Noise, LN – Low Noise, VLN – Very Low Noise, ULN – Ultra Low Noise					

Table 4. Cooling Tower fan-operating conditions in order of decreasing noise.

As one can see from Table 4, the fan speed was lowered to approximately 60% of the base design while the fan shaft power remained fairly constant until Cases 6 and 7. Here the lower efficiencies of the ULN fans significantly increased the required fan shaft power. This is not uncommon for these types of fans as the goal is normally lowest possible noise, not power optimization. It should be noted here that it is highly unusual for a client to accept 300HP motors and in most cases, additional cells or other methods will be employed by the CT OEM to stay at or below 250 HP motors. However, for the purposes of this analysis we have assumed that the client will accept the larger motors.

Fan speed was varied in approximately 10% increments (Base Case to Case 4) by a combination of gear ratios and motor speeds to achieve the required noise reductions. Cases 5, 6 and 7 were the lowest speeds possible for the given CT operating conditions. Multiple fan selections were performed for the STD, LN, VLN and ULN fans types as applicable to achieve the overall most cost effecting solution at that particular speed. Cost increases were estimated for fan Cases 1 through 7 taking into account changes in (if any) fan, motor, gears, mechanical structure and fan stacks. This cost increase estimates were then plotted against the near and far field noise predictions as shown in Figure 9.

The overall CT cost per cell significantly increases as noise is decreased with the lowest noise option increasing the per cell cost by approximately 30%. Given the non-linearity of the data (a general 3rd order polynomial was used to trendline the data) one can see that the cost increase accelerates faster for a relative steady decrease in noise. Put another way, the first 10 dB(A) of far field noise reduction increased the overall cost per CT cell by about 9% from the base design. However the next ***3.4 dB(A)*** increased the cost 28% from the base design.

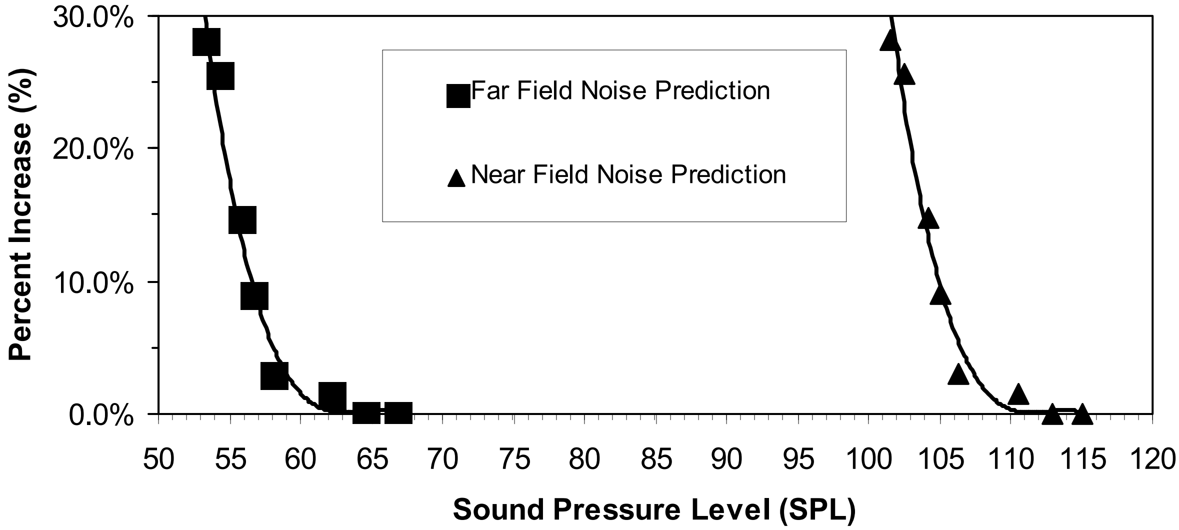

Figure 9. CT cell cost vs. near and far field SPL

The majority of this increase is usually related to the fan and, depending on the type of fan, will range from 20 to 80% of the total increase. This is shown below in Figure 10 where the costs increases are plotted as a function of fan sound power level.

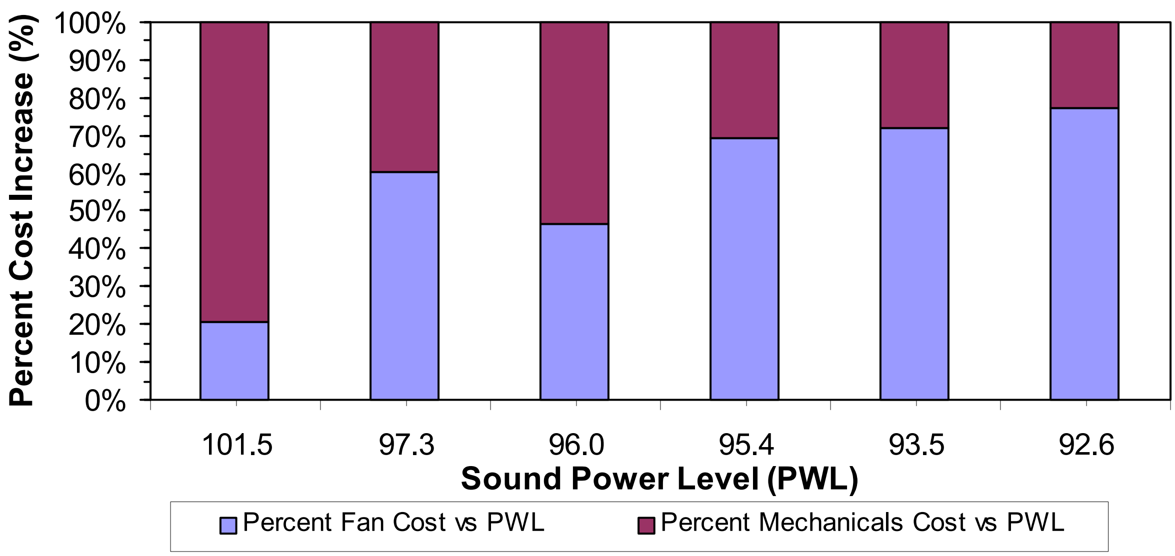

Figure 10. Percent cost increase split between fans and mechanicals

The variance shown in Figure 10 is simply a function of the design points chosen. For example, at the 101.5 dB(A) SPL point, nearly 80% of the cost increase is due to non-fan components due to the fact that the fan cost, in this instance, did not increase that much relative to the speed reduction. In this case, it was the gear driving the cost increase, as it was necessary to move up to the next box size. However, when looked at in total, this combination of fan, motor, gears and so forth provided the least amount of overall cost increase. A good rule of thumb is that the fan will account for approximately 60% of the total cost increase as noise is reduced. However as mentioned in Case 1, every situation is unique and the owner/operator is encouraged to investigate all possibilities with the OEM/Supplier.

SUMMARY

As presented in the two cases herein, the cost of noise, in this case lower noise, significantly impacts the cost of an ACHE or CT depending on how low and where the noise guarantee points are located. And while significant near and far field noise reductions is achievable (in these cases without adding ACHE Bays or CT Cells), as the curves show, the increase in cost is not a linear function but rather a 3rd order polynomial that accelerates quickly as one reduces the noise levels.

While this analysis is far from exhaustive, the authors hope this paper provides the reader with some perspective in the realm of ACHE and CT fan noise. As the old adage goes "there are no free lunches" with the same being true here. However, in this instance, the cost of "lunch" will probably increase rapidly compared to what you receive in return.

REFERENCES

1. American Petroleum Institute, "Air-Cooled Heat Exchangers for General Refinery Service," API Standard 661, Sixth Edition, February 2006.

2. Cooling Technology Institute Standards 111,131, 136, 137 and 153.

3. Cooling Technology Institute ESG 152