Cement Plant Fan Efficiency Upgrades

Allen L. Ray and Derek Couse

***Abstract*—The preheater induced draft (ID) fan, the raw mill fan, and the baghouse ID fan are all major consumers of electrical power in a cement plant. Given such a fierce, competitive marketplace, it is wise, if not imperative, to evaluate these fans and their draft systems to optimize operating efficiency to save input horsepower and, in turn, consumed electrical power. These power-reducing upgrades can also substantially reduce the emissions and the overall carbon footprint of the process. An opportunity arose at a cement plant while the rotor of one of the preheater ID fans was becoming a maintenance issue and reaching the end of its operational life. The foresight of plant personnel reasoned that if it had to be replaced, an upgrade of the rotor should be considered, particularly considering the fact that their electricity provider was offering rebates for power savings. This paper presents a case history including all the challenges along the way to replace the original, antiquated radial blade rotor design with a more efficient, properly selected, and modern backward curved design. The result of this commitment to modernization dramatically lowered the input electrical requirements creating considerable operational savings with an attractive rate on the return of investment.**

***Index Terms*—Backward curved (BC), buildup, cement, efficiency, fans, horsepower, induced draft, mechanical draft, optimize, payback, power savings, preheater, radial, return on investment, right sizing, upgrade.**

I. INTRODUCTION

IN AN industrial process such as cement production, one of the larger parasitic loads is from the horsepower requirements, resulting in electrical power consumption of the mechanical draft fans. Fans that are improperly selected and designed tend to be extremely inefficient, causing increased operational expense, unnecessary carbon emission, continual maintenance issues, and limitations on production. In today's competitive market, it is wise, if not imperative, to examine these fans and the associated draft systems to evaluate optimizing efficiencies and what the return on investment of these power conservation upgrades would yield. These power-reducing upgrades can also substantially reduce emissions and the overall carbon footprint. Within this competitive market, upgrading your fans can be the factor that not only prevents maintenance issues, but essentially provides more production success while eliminating, or at the least minimizing, any forced outages.

Evaluation criteria includes actual operating performance, the cost of power or the cost of generation of power, cost of down time, value of improvements to maintenance issues, allotted payback time, and initial capital investment to perform the upgrade. Fan or draft system modifications can range from correcting inlet or outlet conditions on the fan to a complete rotor redesign and replacement or possibly a complete fan replacement on an existing foundation. Not only is selecting the proper blade design of a fan critical, but matching the fan performance to the system requirements, or "right sizing" the fan is vital for efficient operation. Following all modifications, a post evaluation will be done to determine the actual amount of realized power savings, improved capacity, lower maintenance costs, reduction in carbon emissions, and any additional benefits as well as shortcomings.

A case history on the efficiency optimization of the preheater ID fans at a southern California cement plant that resulted in energy cost reductions of approximately \$0.45 million per year will be presented in this paper.

II. WHY IS EFFICIENCY IMPORTANT?

Today, the market for energy efficiency is being driven by shrinking margins, constant increases in power costs, government mandates at the federal and state levels on reductions of emissions, available tax credits, and incentives offered by utilities. Reducing the cost to produce is a major driving force as clearly a reduction in the amount of electrical power to produce a ton of cement creates room for a price reduction or/and allows for an increase in profit. Mandates to reduce power consumption and increased renewable sources are already being set in place now; it is likely by 2020 that all power consumers will be affected in some way.

Industrial power rates have escalated on average 50% in most areas over the past decade and continue to increase. The current rates vary by region to region with the national average for industrial rates being \$0.067 per kW·hr (Fig. 1). Given the average rate and 8000 h of operation annually, each horsepower consumed per year costs \$400 ($8000 \times 0.067 \times 0.7457$).

Of course, with any business decision to spend capital, there needs to be an attractive rate on return on investment for any conservation upgrade. Also, an intangible benefit of these power-reducing upgrades is the reduction of carbon emissions. Finding candidates involves an evaluation process that includes gathering actual performance data, determining the cost of power, the cost of down time, the value of improvements or potential chal-

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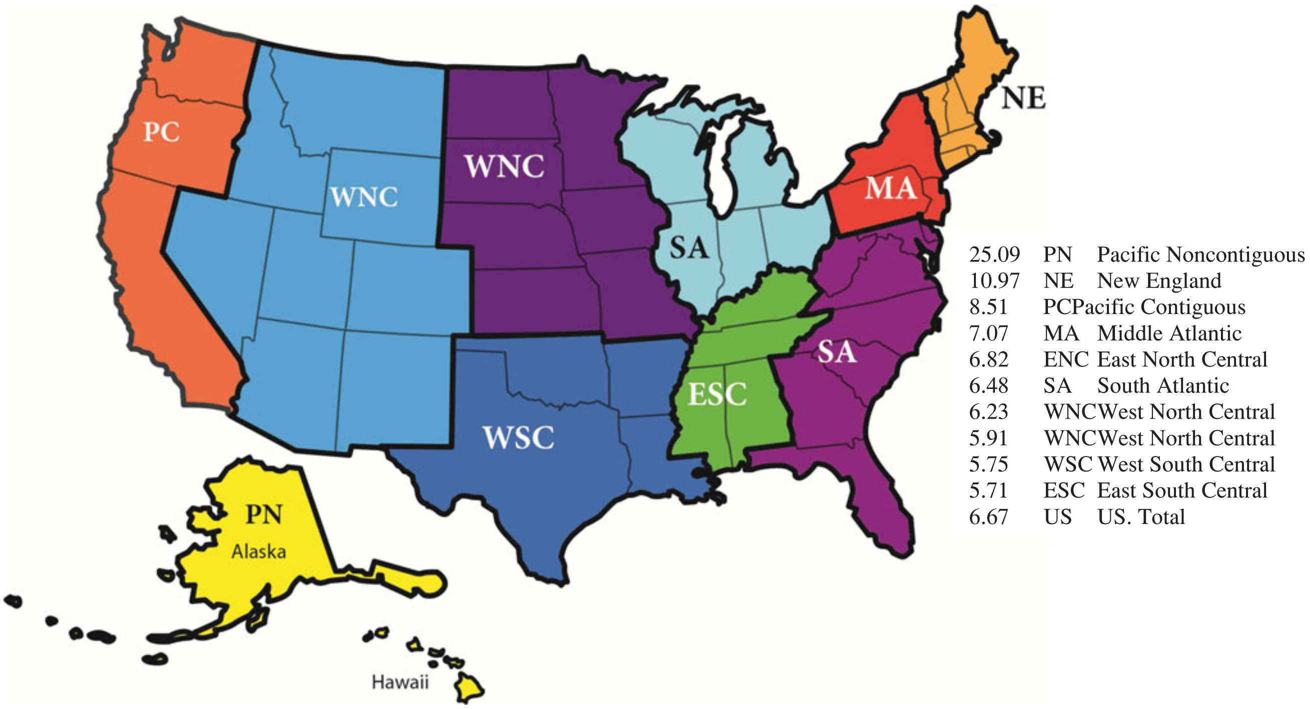


25.09	PN	Pacific Noncontiguous
10.97	NE	New England
8.51	PCPacific Contiguous	
7.07	MA	Middle Atlantic
6.82	ENC	East North Central
6.48	SA	South Atlantic
6.23	WNCWest North Central	
5.91	WNCWest North Central	
5.75	WSC	West South Central
5.71	ESC	East South Central
6.67	US	US. Total

Fig. 1. Electrical rates by region (US$ per kW·hr).

lenges of maintenance issues, allotted payback time, and initial capital investment to make these improvements.

Mechanical draft fans have a “sweet spot” where they operate at maximum efficiency. It is desirable to accurately size a fan with a proper width to diameter ratio for a given set of performance conditions (volumetric flowrate, static pressure, and inlet density) to operate at that spot. The peak efficiency point on their fan curve is normally located just to the right of the peak of the capacity curve. A fan engineer will normally size and design the fan such that the performance curve and the system resistance curve intersect at the most efficient and stable point. Also, if the system resistance curve is not accurately predicted, this “sweet spot” of operation will likely be missed and performance will suffer. Therefore, one can easily see the importance of correcting, predicting, or measuring the pressure requirements of the system (system resistance) and “right sizing” the fan equipment.

Existing equipment offers a golden opportunity to “right size” the fan equipment when it becomes time for an operational “end of life” equipment replacement. The actual system resistance can be accurately determined by field measurement and, in turn, an appropriately sized and designed fan can be selected. This type of replacement will not violate any of the regulations or permitting in place as it does not change the performance of the equipment, i.e., the flowrate, but only lowers the amount of power consumed.

III. PROCESS OF THE PREHEATER ID FAN UPGRADE

A southern California cement plant did a preliminary investigation in 2002 and discovered that their preheater induced draft fans were operating inefficiently. Initial field performance testing conducted in October 2003 showed the fans to be operating at approximately at 55% static efficiency (Es) in normal production even when operating using speed control. It was determined in the early investigations that the system resistance required was lower than originally designed for, that is, a lower required pressure for a given volumetric flowrate. The plant had made modifications along the way in the preheat tower duct system lowering the overall system losses. This changed the point of rating on the characteristic curve allowing for the fan to operate now at a higher volumetric flow rate for increased production albeit at a lower Es (Fig. 2).

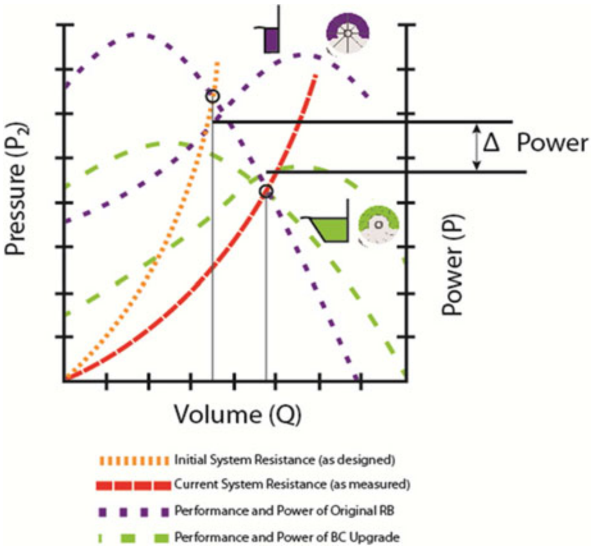


Fig. 2. Graphic representation of performance of original rotor to upgrade rotor.

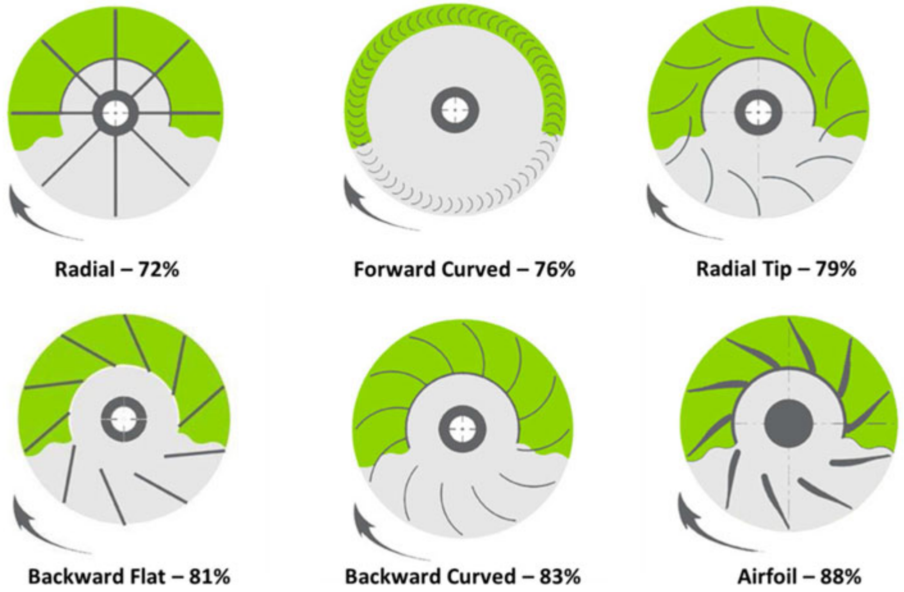
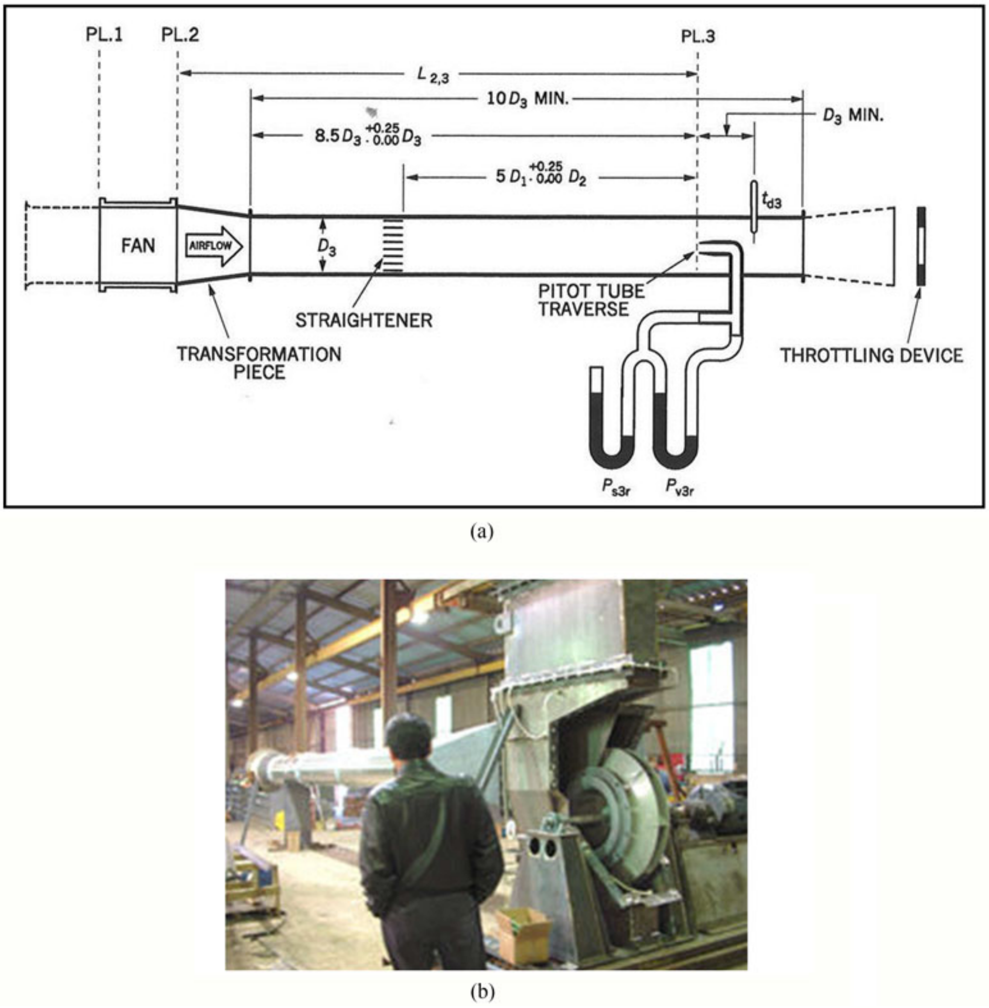


Fig. 3. Standard blade shapes commonly used.



(a)

(b)

Fig. 4. (a) AMCA 210-07 Fig. 7A schematic for performance test apparatus. (b) Test apparatus for physical model performance testing.

Today, most fan original equipment manufactures (OEM's) build (Fig. 3) six different blade styles or minor variations of these designs. A fan designer will attempt to match the blade style that is the most efficient but still suitable for the service environment. For a preheater application, the most efficient design would be a backward curved (BC). The only design more efficient would be a full airfoil blade but given the possibility of some minor erosion, the decision was made to upgrade to a BC. To minimize the costs of this upgrade as well as allow for a fallback position, it was decided to attempt to retrofit the BC rotors into the existing radial blade (RB) housings. New inlet cones would also be designed and fabricated to mate the upgraded rotors into the existing housings. Fitting the upgraded rotor into the existing housing gave comfort to the plant that it could return to a fan system that was proven, should the new approach fall short of their expectations and allow them to fulfill their primary objective to make cement.

The RB housings had very narrow scrolls as compared to the normal housings of the BC designs being considered. The main challenge became to develop an inlet cone or bell that would allow the 90° turn in the centrifugal rotor without flow separation given the short distance to work with. Also, the BC rotor would be larger than the RB rotor because the pressure generating capability as a function of the diameter was lower for the BC design versus the RB design.

Extensive analysis and engineering was done to insure that the BC rotors would perform in the existing housings. Prototype model testing was conducted putting a scaled model of the proposed BC rotors into a corresponding scaled model of the existing RB housings (Fig. 4). This testing was done in accordance with AMCA 210-07: laboratory methods of testing fans for certified aerodynamic performance rating using a Figure 7A: outlet duct setup—pitot traverse in outlet duct with cell straightener.

A characteristic curve is developed by measuring the volumetric flowrate, static pressure, inlet density, and brake horsepower at set of points from zero delivery to full delivery by throttling the outlet of the test apparatus from completely closed to full open

$$\mathrm{BHP} = \frac{(Q \ \times \mathrm{Ps} \times \mathrm{Kp})}{6362 \ \times \mathrm{Es}}$$

where Q is the flowrate (acfm—actual cubic ft. per minute), Ps is the static pressure (inches of water column), Kp is the compressibility, Es is the static efficiency, and BHP is the brake horsepower (horse power (HP) at fan shaft) (Fig. 5).

After several iterations of changing inlet cone geometry, the model test predicted an Es of +76.0%. By applying the basic affinity laws of fans, (Fig. 6) the performance of the prototype test was scaled up to deliver the measured flowrate and pressure at the normal production rate

$$Q_2 = Q_1 \ \times \ (D_2/D_1)^3 \times \ (N_2/N_1)$$

$$\mathrm{Ps}_2 \ = \mathrm{Ps}_1 \ \times \ (D_2/D_1)^2 \times \ (N_2/N_1)^2 \ \times \ (\rho_2/\rho_1)$$

$$\mathscr{P}_2 \ = \mathscr{P}_1 \ \times \ (D_2/D_1)^5 \times \ (N_2/N_1)^3 \ \times \ (\rho_2/\rho_1)$$

where Q is the flowrate, D is the diameter, N is the speed, Ps is the static pressure, ρ is the density, and $\mathscr{P}$ is the BHP.

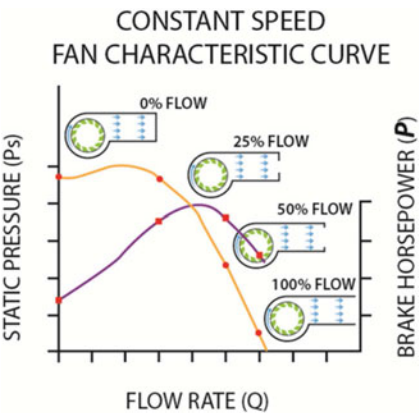


Fig. 5. Developing a fan performance curve.

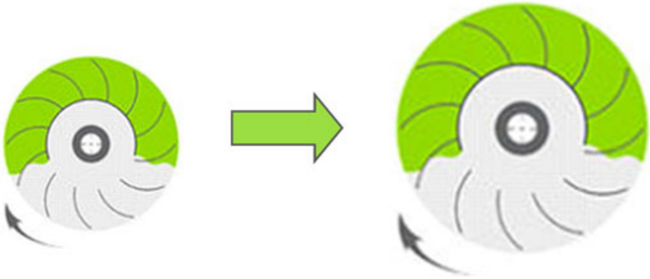
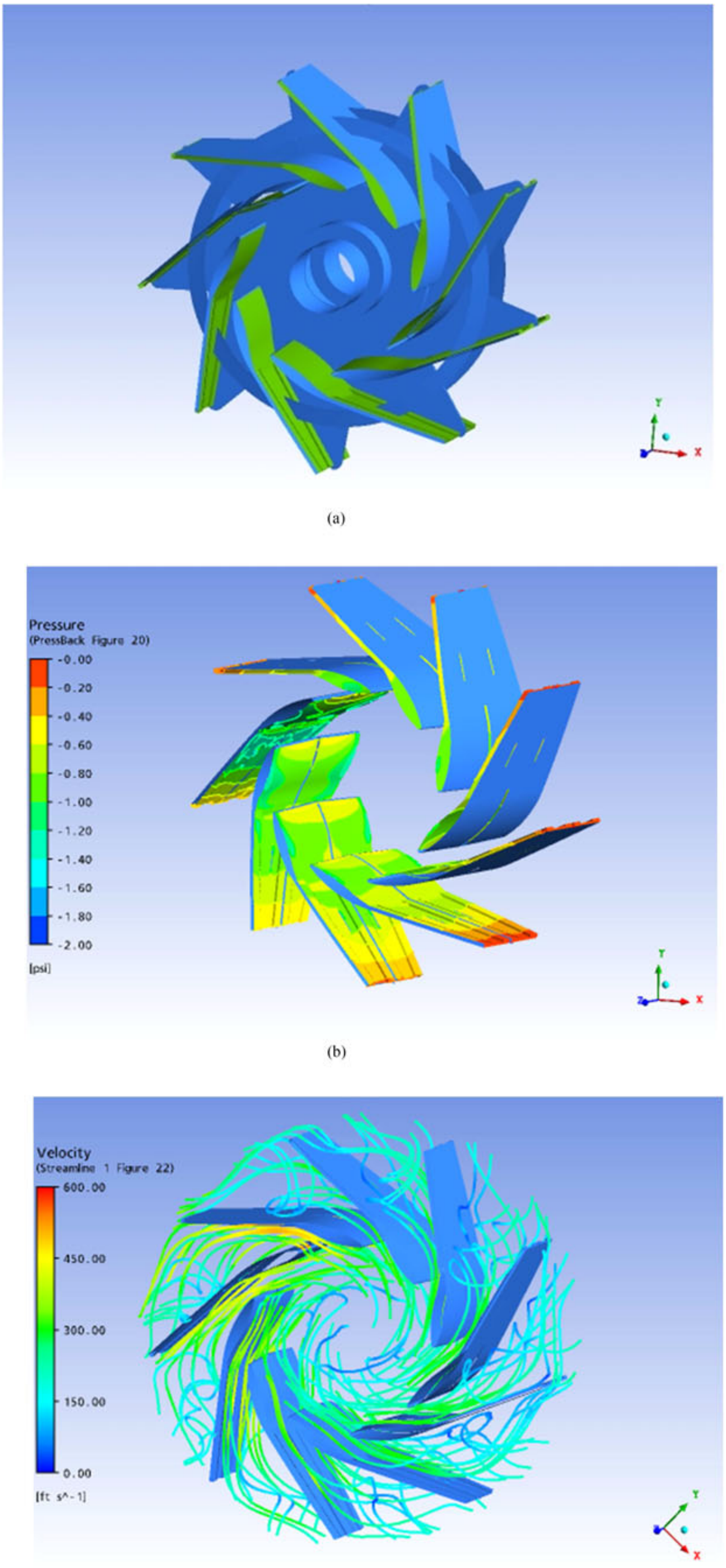


Fig. 6. Scaling from prototype to operational size rotor.

TABLE I
SUMMARY OF THE METRICS OF THE UPGRADE

Field measured	Original design	Upgraded design
Q = 329 131 acfm per fan	105.625 in effective diameter	117.75 in effective diameter
Ps = 31.10 in WC	RB design	BC blade
Power = 2332 BHP	Installed in the early 1980s	Installed in existing housing
	Q = 275 000 acfm ea. fan	Q = 329 131 acfm ea. fan
	Ps = 40.0 in WC	Ps = 31.1 in WC
	Power = 2254 BHP	Input power = 1912 BHP ea. Power decrease = 420 BHP ea Electric rate = \$0.082/kW·hr Savings = US\$536/HP/year ea. Total savings/year = \$225 120 Project total costs = \$194 925 Time of ROI = 316 d CO_2 reduction = 8000 t/year

The production rotors were installed in two separate phases with the first rotor being put into service in August 2007. The second rotor was recently installed in late fall of 2014. Both rotors, at a production rate of 340 ton/h, saved from 700 to 800 hp (522–597 kW). At \$0.082 per kW·hr cost, the realized savings are US \$536 per HP per year equating to annual savings of \$375 200–\$428 800 annually. In summary, the predictions and promises of power consumption were met, if not exceeded (Table I).



(c)

Fig. 7. (a) Solid model illustrating blade shape. (b) Mapping of pressure distribution on back side of the blades. (c) Velocity streamlines on new blade shape.

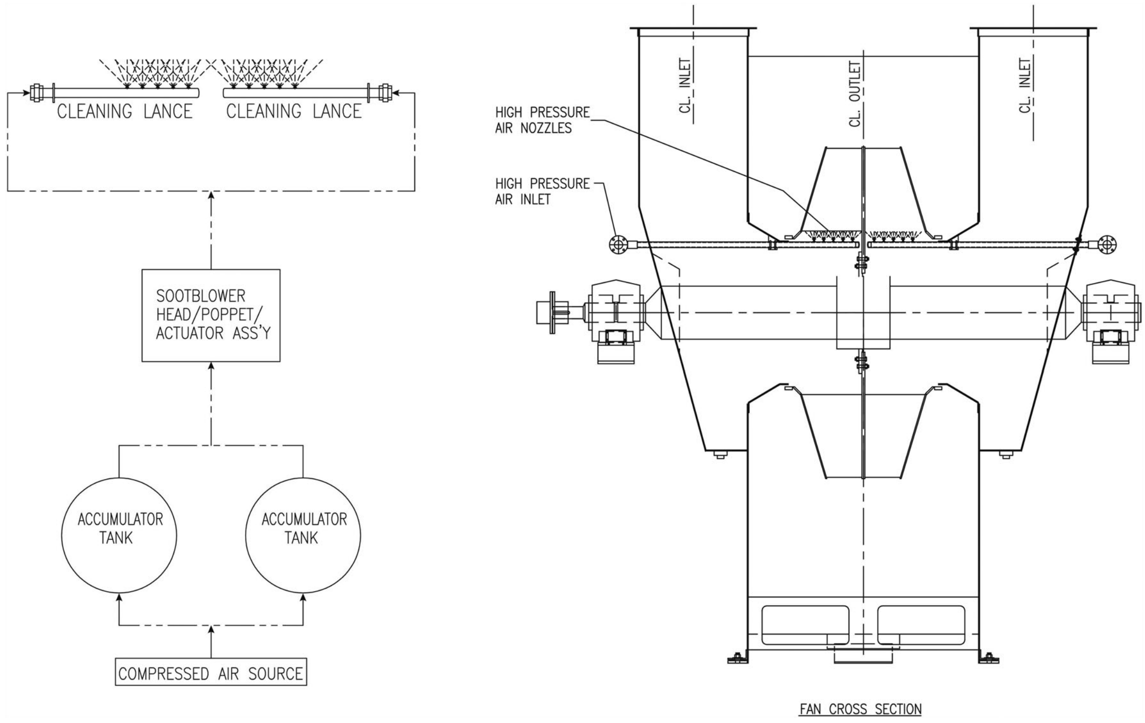


Fig. 8. Compressed air cleaning system.

IV. CHALLENGES AND SOLUTIONS

During the initial planning stage, there were concerns that there could be some minor buildup issues with carryover product collecting on the back side of the BC blade. The RB design was chosen in the 1980s because of its advantage of being the best design to self-clean and prevent these buildup issues. However, the BC blade design has the potential to collect material on the backside of the blade; this area has a concave shape where laminar flow will separate away from the surface leaving a boundary layer with little to no flow allowing material to settle and deposit. Also, this material collection is aided by the centrifugal forces pinning and trapping the materials to the blade backside as well.

The desire to save energy was enough incentive to go forward with the BC design for the rotors. Getting the first rotor into service was not without some growing pain. When the first rotor was put into service in August 2007, the rotor experienced severe vibration problems due to excessive material accumulation on the back of the blades. The material was accumulating in the region between the inner edge of the blade and the stiffener installed on the backside of the blade for structural reasons; after the accumulation built up to a certain thickness, uneven portions of the material would break away, causing unbalance and vibration. The material was not adhering to the fan blades, but was held in place by centrifugal force; once the fan speed was lowered, the material would all fall off. It was generally agreed that the stiffener as designed acted like a dam in the flow along the backside of the blade and allowed the problematic material build up. Given the low inventory of clinker at the time, the plant needing to operate and the time to remedy longer than could be tolerated; it was decided to remove the BC rotor and return the RB rotor into service until a solution could be determined.

Considerable effort was poured into this setback. Using a current version of a computational fluid dynamics (CFD) program, models were built to analyze options on how to eliminate the culprit blade stiffener that was suspected of causing the buildup on the backside of the blades. In the end, the decision was made to use an airfoil-shaped nose bar on the leading edge of the blade; this change offered:

1) stiffness needed to keep the stresses in acceptable limits from the centrifugal force;
2) elimination of the dam effect from the first stiffener with the CFD model predicting a huge reduction in buildup potential on the blade backside;
3) improvement in the efficiency.

The CFD data shows that the design change has significantly improved the material accumulation problem. The performance change indicates that the airfoil nose has lowered the pressure producing capability by 5.4% and increased the efficiency by 6.3%. This means that the fan will need to run slightly faster to produce the needed static pressure, but it will produce it at a lower power cost. Another indication of the improvements related to the material accumulation is the change in the pressure profile on the back of the blades (Fig. 7).

The experience of our CFD consultants has been that the material accumulation will be greater in the regions of low pressure. The first BC rotor was modified and then put back into service. Although the buildup was greatly diminished, there were still problems with unbalance from buildup, particularly at startup. A compressed air lance system was installed on inlets of the BC rotor (Fig. 8).

Fig. 9. Photos of upgraded rotor.

The system included a poppet valve and large accumulator tanks. On set intervals, the poppet valve will open, allowing a large volume of compressed air to “sweep off” or clean the backside of the blade. With the redesigned stiffener and the addition of the air cleaning system, the unbalance and vibration issues can be managed on these new rotors.

V. SUMMARY

Upgrading the preheater fans was not without challenges and setbacks. It took the resolve and the ingenuity of both the fan supplier and the cement plant working through these issues to make this endeavor a successful one. In the end, the cement plant has an efficient rotor design that reliably operates, provided a short period to return on investment when including the incentives from the electrical power provider and will continue to pay back for many years to come (Fig. 9).

Projects such as these are not only beneficial to the company doing it, but they also benefit the immediate community, the state, and even the nation. This reduction in electrical consumption gives back electrical capacity to future customers without any capital investment to the electrical power producer and supplier. It also reduces impact on the environment by lowering greenhouse gases as well as conserving natural resources for future generations. In closing, a project such as this is a win-win for everyone. It sets an example for others to follow as there are many more opportunities to reduce energy consumption, save money, protect the environment, and conserve natural resources.

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