

Lucerne Valley's ID fan upgrade

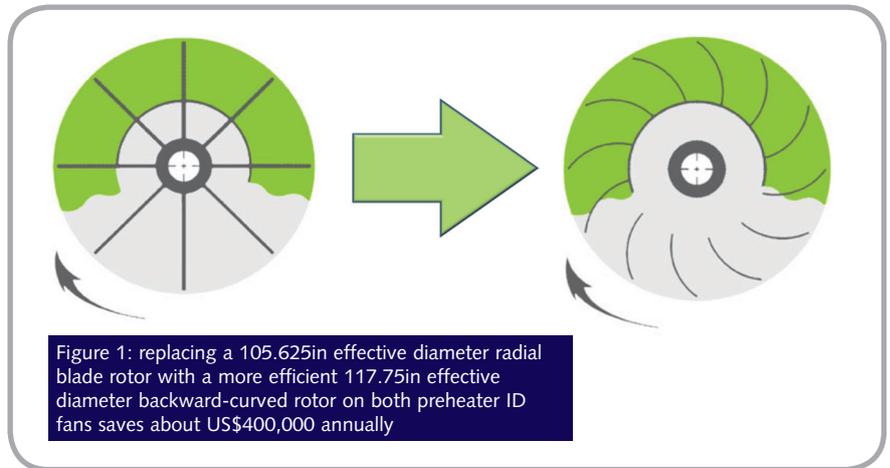
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Of course, with any business decision to invest, there needs to be an attractive return on investment (ROI) for any conservation upgrade. An intangible benefit of these power-reducing upgrades is the reduction of carbon emissions. Finding candidates involves an evaluation process that includes gathering actual performance data, determining the cost of power, the cost of downtime, the value of improvements or potential challenges of maintenance issues, allotted payback time and initial capital investment to make these improvements.

Mechanical draught 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 the fan curve is normally located just to the right of the peak of the capacity curve. A fan engineer will usually 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 correctly predicting or measuring the pressure requirements of the system (system resistance) and 'right-sizing' the fan equipment.

Mitsubishi Cement in Lucerne Valley, California, carried out a preliminary investigation in 2002 and discovered that its preheater or kiln ID fans were operating inefficiently. Initial field performance testing conducted in October 2003 showed the fans to be operating at

The preheater induced draught (ID) fan, the raw mill fan and the baghouse ID fan all are 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 draught systems to optimise their efficiency saving input horsepower and, in turn, consumed electrical power. This article reports on Mitsubishi Cement's need to optimise its kiln ID fan at the Lucerne Valley plant, USA.



Mitsubishi Cement Lucerne Valley plant ID fan

Original design

- $\phi 105.625$ in effective diameter
- radial blade (RB) design
- installed in the early 1980s
- $Q = 275,000$ acfm ea. fan
- $P_s = 40.0$ in WC
- Power = 2254bhp

Field measured

- $Q = 329,131$ acfm each fan
- $P_{sr} = 31.10$ in WC
- Power = 2332bhp

approximately 55 per cent static efficiency 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. The plant had made modifications along the way in the preheater 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 lower static pressure and brakehorse power (bhp) albeit at a lower static efficiency E_s (see Figure 1).

$$E_s = (Q \times P_s \times K_p) / 6362 \times \text{bhp}$$

Q = flowrate (acfm – actual cubic ft/min)

P = static pressure (in WC)

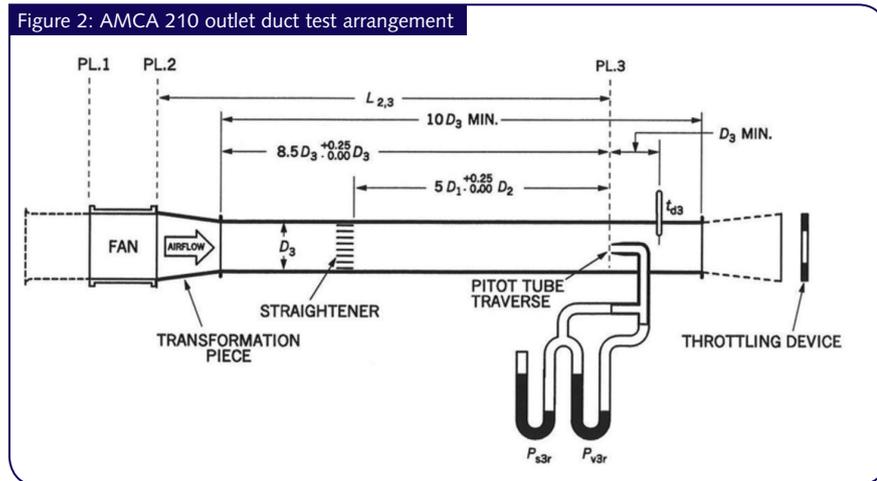
K_p = compressibility

E_s = static efficiency

Equation 1: Fan Static Efficiency – E_s

Also, it was understood that the existing fans were radial blade designs that had been installed in 1980 and were not the most efficient fan that would likely be applied today. A plan was devised to not only 'right-size' the rotors by re-designing to get back to that 'sweet spot' but also to upgrade the rotors from radial blade (RB) designs to backward curved (BC) blades, which would be inherently more efficient (see Figure 1).

To minimise the costs of this upgrade,



it was decided to attempt to retrofit new BC rotors into the existing RB housings. New inlet cones would also be made that would mate the new rotors to the existing housings. 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. In addition, 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 carried out to ensure 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. This testing was performed in accordance with AMCA 210 using an outlet duct arrangement (see Figure 2).

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. After several iterations of changing inlet cone

geometry, an optimum E_s of +76 per cent was measured in the model test.

By applying the basic affinity laws

for fans, the performance of the prototype test was scaled up to deliver the measured flowrate and pressure at the normal production rate.

The production rotors were installed in two separate phases with the first rotor being put into service in August 2007. The second rotor was

Affinity Laws for fans

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

$$P_{s2} = P_{s1} \times (D_2/D_1)^2 \times (N_2/N_1)^2 \times (r_2/r_1)$$

$$P_b = P_1 \times (D_2/D_1)^5 \times (N_2/N_1)^3 \times (r_2/r_1)$$

Q = flow rate | D = diameter | N = speed
 P_s = static pressure | ρ = density
 r = brake horsepower

Upgraded fan design

ϕ 117.7in effective diameter
 backward curved blade
 installed in existing housing
 $Q = 329,131$ acfm each fan
 $P_s = 31.1$ in WC
 input power = 1912bhp each
 power decrease = 420bhp each
 electric rate = US\$0.082/kWh
 savings = US\$536/hp/year each
 total savings/year = US\$225,120
 project total costs = US\$194,925
 time of ROI = 316 days
 CO_2 reduction = 8000tpa

recently installed in late fall of 2014. Both rotors, at a production rate of 340tph, save from 700-800hp (522-597kW). At US\$0.082/kWh cost, the realised savings are US\$536/hp/annum equating to total annual savings of US\$375,200-428,800. In summary, the predictions and promises of power consumption were met, if not exceeded.

During the initial planning stage, there were concerns that there could be some minor build-up issues with carry-over product collecting on the back side of the backward curved blade. The radial blade was chosen in the 1980s because of its advantage of being the best design to self-clean and prevent build-up issues.

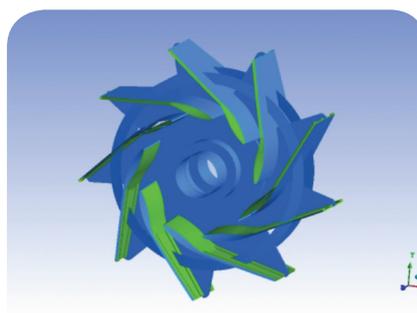


Figure 3: airfoil nose and ring stiffeners

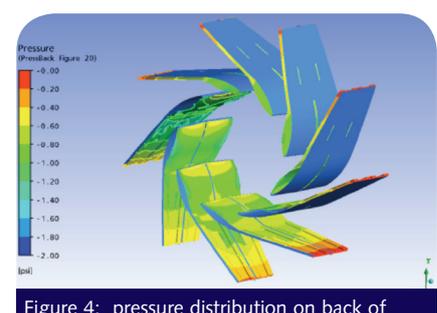


Figure 4: pressure distribution on back of blade

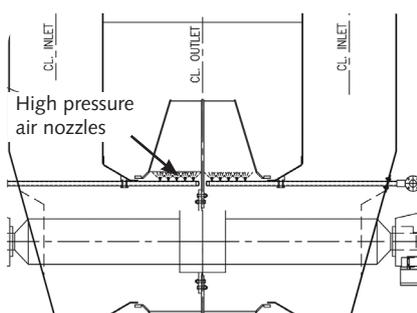


Figure 5: fan cross-section illustrating cleaning system with highpressure spray nozzles



Figure 6: build-up that causes problems with unbalance, particularly at start-up

Figure 7: ID fan upgraded rotor installation



On the other hand, the backward curved 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.

However, the desire to save energy was enough of an 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 commissioned in August 2007, it experienced severe vibration problems due to excessive material accumulation on the back of the blades. 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

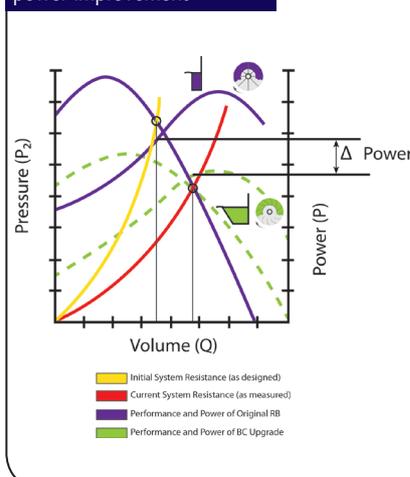
material build-up. Given the low inventory of clinker at the time, the plant needing to operate and the time to remedy longer than could be tolerated, it was decided to remove the BC rotor and return the RB rotor into service until a solution could be determined.

Considerable effort was poured into this setback. Using ANSYS CFX, computational fluid dynamic (CFD) models were built to analyse options on how to eliminate the culprit blade stiffener that was suspected of causing the build-up on the back 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. The rigidity needed to keep the stresses from the centrifugal force within acceptable limits.
2. It eliminated the dam effect from the first stiffener with the CFD model predicting a huge reduction in buildup potential on the back of the blade.
3. It showed an efficiency improvement.

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 per cent and increased the efficiency by 6.3 per cent. 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. The experience of our ANSYS consultants has been that the material accumulation will

Figure 8: graph illustrating power improvement



be greater in the regions of low pressure (see Figures 3-6).

The first BC rotor was modified and then put back into service. Although the build-up was greatly diminished, there were still problems with unbalance from build-up, particularly at start-up.

A compressed-air lance system was installed on the BC rotor (see Figure 5). 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 back 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.

In closing, upgrading the preheater fans was not without challenges and setbacks. However, in the end, Mitsubishi Cement has an efficient rotor that reliably operates and will continue to pay back for years to come.

Figure 9: preheater ID fan upgraded rotor

