TEMPORARY REPAIR OF A LARGE AXIAL BLAST FURNACE MOTOR-BLOWER AS A SOLUTION FOR A STEEL MILL OPERATIONAL CONTINUITY

by

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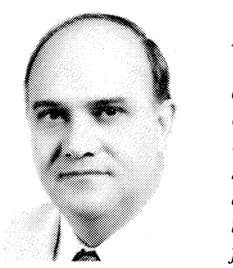

Luiz Fernando S. Mendes, President of Promentec, Ltd., founded the company in 1972 as a maintenance engineering and consulting company in the field of turbomachinery, covering the needs of the petrochemical, chemical, siderurgy, paper, and sugar companies. Vibration analysis, balancing, and machinery repair works were incorporated into the company activities following customer demands over the years.

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ABSTRACT

The repair work of a large axial blower that had its rotor and stator blades severely damaged is described. The 43 MW axial air blower is of critical importance for a large steel mill in Brazil, and the repair time was of the utmost importance. The circumstances in which the failure occurred, along with the investigation work for the causes, are also discussed. The failure of a second blower while the first one was under repair, which brought about a serious risk of loss of a blast furnace, and the measures taken to put it back to operation in a very short time, are also discussed.

The purpose herein is to demonstrate the feasibility of reliable temporary repairs on mechanically damaged machines to reduce production loss in a very short time.

INTRODUCTION

One of the largest and most important steel mills in Brazil, the Brazilian National Steel Company (CSN), has three blast furnaces fed by two identical axial motor-blowers, commissioned in May 1976. Three older turboblowers are normally kept only as an emergency standby, because of their low capacity. The plant was operating at full load with the Number 1 motor-blower supplying air to blast furnaces 1 and 2, and the Number 2 blower to the largest capacity blast furnace 3 (Figure 1).

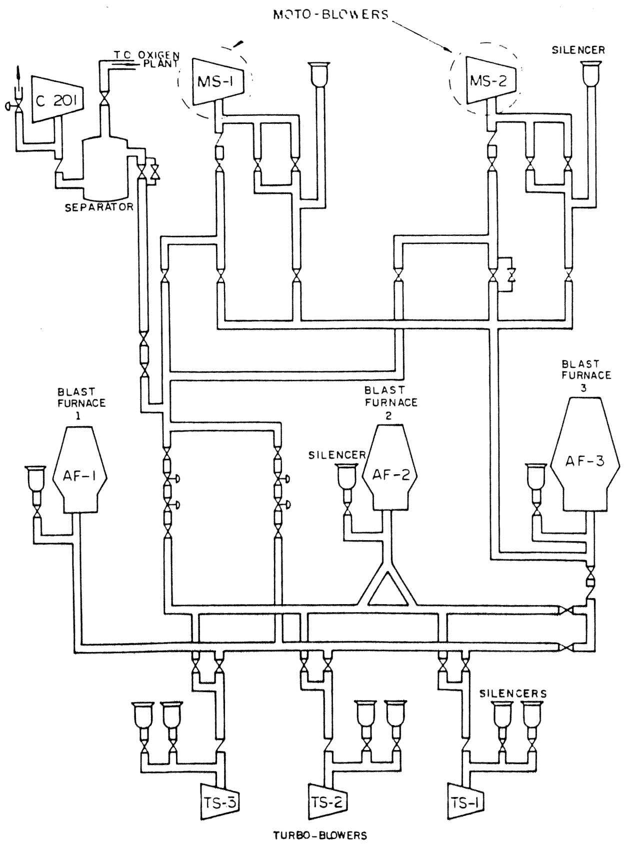

Figure 1. Simplified Flow Diagram of Motor-Blowers Arrangement to Feed Blast Furnaces 1, 2, and 3.

The capacity of each motor-blower is 7200 m3/min (42,400 scfm) at 4.95 atg (70.4 psig), with a power consumption of 43 MW. The blower rotor has 13 blade rows, and there are 13 adjustable stator blades with one additional row of adjustable inlet guide vanes (Figure 2). The blower rotor weight is 26.5 metric ton (58,300 lb), with a total length of 7.8 meters (25.6 ft), a distance between bearing centerlines of 6.2 m (20.34 ft), and a maximum diameter of about 1.5 m (59 in).

THE ACCIDENT

In June 1991, the Number 2 motor-blower was manually tripped down by the operator, because an unusually high vibration and noise level was suddenly noticed. Later, internal inspection revealed that three first stage and three second stage rotor blades had broken and damaged all other rotor blades, along with the stator blades (Figures 3 and 4). The resulting impacts also damaged the mechanical linkages for the stator blades and their guide sleeves and bushings.

It should be emphasized that all maintenance work in these motor-blowers was done under the supervision of the manufacturer. Actually, the last preventive maintenance of both blowers took place from October to December 1990. Despite the maintenance, the failure happened soon afterwards.

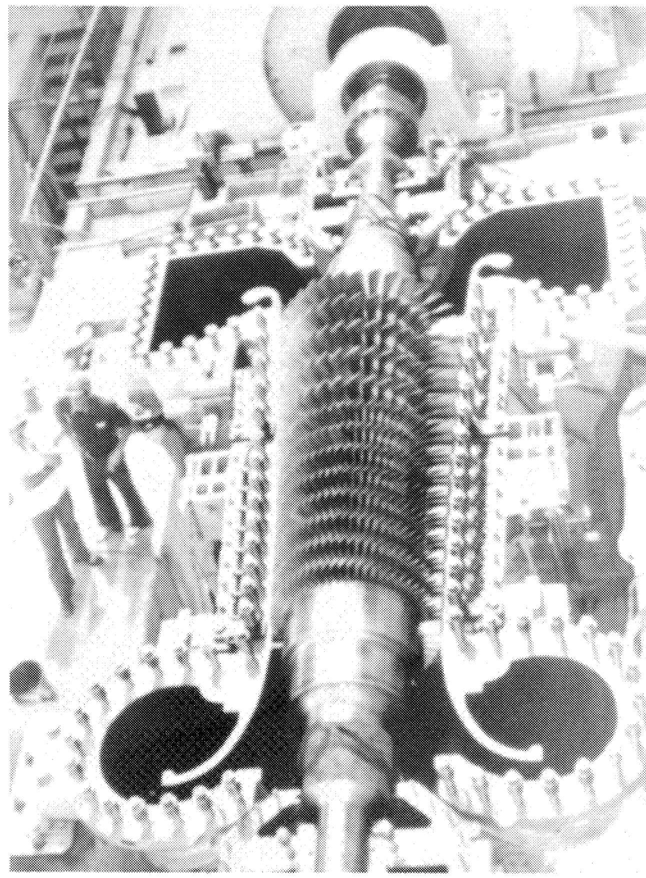

Figure 2. View of Open Blower Showing Blade Arrangement.

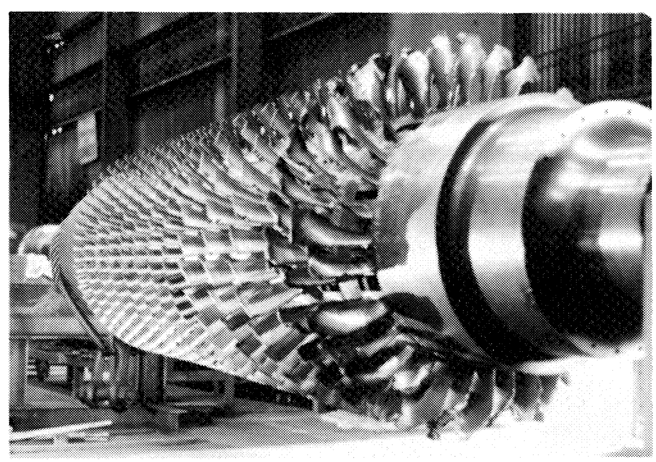

Figure 3. View of Damaged Rotor Blades.

Inspection of the broken blade roots indicated fatigue crack propagation and visual signs of fretting corrosion. This fact also brought a concern about the other Number 1 motor-blower that could be in similar condition, as the cumulated operating time was about the same for both blowers.

THE DECISION TO REPAIR

The first contact made by CSN was with the blower manufacturer, who proposed to resolve the problem by removing all rotor and stator blades and replacing them with new ones. The rotor would

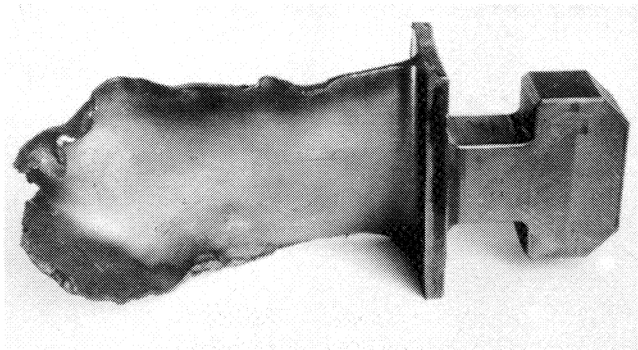

Figure 4. View of a Damaged Airfoil of a First Stage Blade.

have to be sent back to their factory abroad for a detailed investigation of the causes of the problem. The stator would be repaired onsite, and all affected parts of the blade's driving mechanism would be replaced. The total estimated repair time was about 10 months.

Other measures taken by CSN involved the investigation of the existence of another identical axial compressor for spare parts or the purchase and installation of a new, or even used, axial compressor (if available) that could fit in the base with minor changes. The adaptation of a different compressor could not be made in less than seven months.

In view of the urgency dictated by high production loss and the potential risk of a similar failure on the Number 1 blower, the proposal was made to make a temporary repair eliminating the first and second blade rows that were most severely damaged and to repair the other less damaged blades. Only a few spare blades were available in stock at the plant (six for each rotor stage and about 25 for each stator stage). The casing had only minor damages and could be repaired onsite. The stator blade guide sleeves, along with the driving linkages, could be manufactured locally, and the guide bushings could all be obtained in short time as standard parts. All this would make it possible to return the blower into operation in three to four months instead of the 10 months for a complete repair proposed by the manufacturer.

Meanwhile, CSN was also proceeding with a plan to install a pipeline (1,450 m long and 0.61 m in diameter) to send air from their air compressor at the oxygen plant to blast furnaces 1 or 2. This work was estimated to take about two months.

At this point, the risk of air shortage could not only affect production but could also lead to a complete loss of the blast furnace Number 3 in case of an accident in the other motor-blower still in operation. Unfortunately, this problem eventually occurred, as will be discussed later.

CHANGE OF PERFORMANCE CHARACTERISTICS

It was necessary to demonstrate to the steel company that even with the reduction in capacity (with only 11 stages instead of the original 13 stages), the blower could still meet their production needs. The original blower performance curves with 13 stages were duplicated by computer simulation to determine the basic parameters [1], and a new set of curves were generated using the parameters for the remaining 11 stages. The original surge limits were retraced, and a new operating range was determined (Figure 5). Meanwhile, CSN determined their minimum needs of air flow and pressure to attend the production with blast furnaces 1 and 2. The corresponding point was within the new operating range with enough margin for surge protection as indicated. In this way, it was quite sure that the existing settings of the surge protection system would not have to be altered.

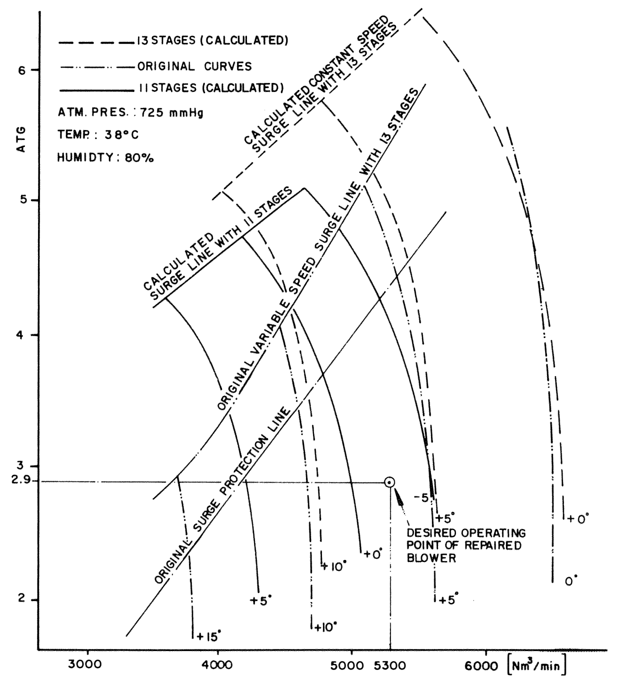

Figure 5. Original (13 Stages) and Modified (11 Stages) Performance Curves.

STUDY OF THE CAUSES OF THE FAILURE

On the first visit to the plant for a visual inspection of the blower after the accident, clear signs were observed of fretting on the "neck" of the inverted "T" type root (Figure 6) of broken blades in their contact surfaces with the rotor grooves. Those surfaces presented a thin layer of a typical reddish brown color oxide. All blades generally were found loose in their grooves, and they could easily be moved by hand [2]. Although the centrifugal force in normal operation would keep the blades forced against the "shoulders" of the inverted "T" grooves, the freedom of the blades to make small movements within the existing clearances between them was the most likely cause of fretting.

Fretting corrosion was confirmed in a study made by a specialized research institute [3]. All cracks initiated on those contact

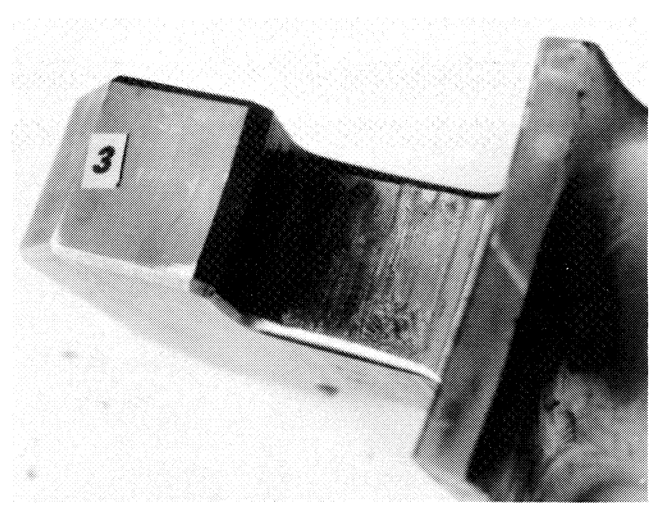

Figure 6. View of a First Stage Blade Root Showing Fretting Corrosion.

surfaces and propagated transversely across the root neck (see Figure 7). Micrographic analysis also indicated that several other microcracks existed on those surfaces.

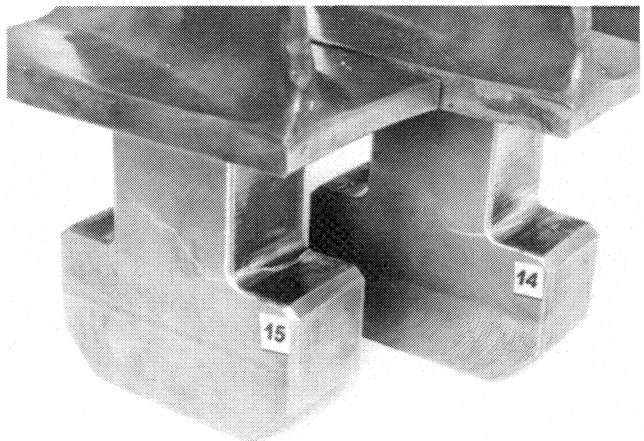

Figure 7. View of Cracks in Blade 14 of First Stage and Blade 15 of Second Stage.

The fretting that resulted was most likely due to a process in which the initial blade close clearance was slowly increasing with time, allowing the increase in the relative movement. The crack propagation was then due to fatigue process caused by the combination of high static centrifugal load and the variable stresses due to blade vibration. The blade root surfaces were showing a poor finishing, as there were clear machining marks and irregular surface roughness in the original blades (Figure 8), and this problem certainly made conditions favorable for fretting to occur. The spare blades in stock had been recently ordered by the steel company, and the surface finish of the root surfaces was superior, when compared to the original blades.

Samples of deposited material were removed from the rotor grooves, but their analysis indicated no contaminants that could be related to the cracks.

THE REPAIR WORK

Preparation

At first, a detailed inspection was made of each of the rotor blades to evaluate the possibility of repair. The rotor blades were identified and classified according to their condition: blades that could be repaired, blades that could not be totally repaired, but could be cut at the tip to reduce their height to be used on one of the downstream stages, blades that were not reparable at all due to excessive damage, and a few blades that were in good condition and could be used "as is." The first five rows of blades were all of the same profile, with decreasing heights from first to fifth stage. The blades of the remaining stages were smaller, but were also all of the same profile, with decreasing heights from the sixth to the thirteenth stage. This classification work was necessary to anticipate the extension of the work, in order to predict the time necessary for the blade repair and to determine that it would actually be possible to restore all 11 stages as expected. The stator blades were in much better condition, and the number of spare blades available in the plant made the work easier than the rotor blades.

The rotor blades were identified with a number indicating its stage and its individual position in the groove. After disassembly, each blade identification number was punched at the bottom end of its root for reference.

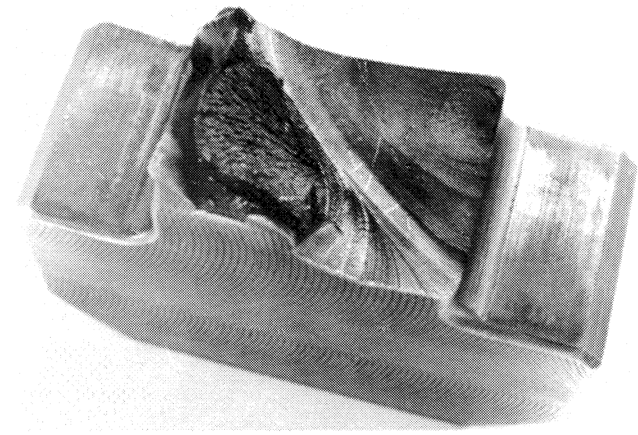

Figure 8. Machining Marks in a First Stage Blade Root and Typical Fractured Neck.

The disassembly of the blades presented some difficulties because some blade roots were distorted because of the very high stress level at the time they were broken and forced against the groove surfaces. The grooves also had some small burrs and deposits of corrosion products (rust).

After removing all blades, a detailed dye penetrant and magnetic particles inspection was made of the grooves, and a dimensional inspection was made with the shaft supported on V-blocks to check its eccentricity with respect to the journals. The results were always within acceptable limits (maximum of 30 micrometers or 1.2 mils total indicator reading at the middle of the shaft), even though the figures were influenced by slight thermal bows in each time it was measured. This is typical for very large and massive rotors.

All blades were subjected to magnetic particle and dye penetrant tests to ensure that no cracked blades would be repaired. Eight additional blades other than those broken ones were found with cracks on the root and were rejected.

Rotor Blades Repair

Damaged blades were first mechanically straightened and the original profiles were manually restored by trained personnel, using templates specially prepared, based on the shape of new blades (Figure 9). In particular, the smoothness of the curvatures and the angles of leading and trailing edges were carefully controlled to keep them as close as possible to their original values. To make this possible, a "production line" was prepared, and each individual blade was selected and repaired in accordance with the classification previously made to avoid working on excessively damaged ones.

The material hardness on the restored blades was controlled to ensure that the mechanical work had not affected the ductility of the blades. The values on the original surface were in the range of 22-26 Rockwell C in one of the blades and 34-38 in another blade (two different values found on two different original blades), and in repaired areas, the values were between 24-26 in the first case, and 34-36 Rockwell C in the second case, indicating that there was no problem of surface hardening. This was of utmost importance in order to avoid risk of cracks originating from localized stresses.

The repaired and new spare blades prepared for each of the rotor stages were carefully weighed to a tolerance of one gram, and groups of blades were made to arrange equal weight blades in opposite sides on the mounting grooves in order to minimize final unbalance due to weight distribution around the rotor (Figure 10).

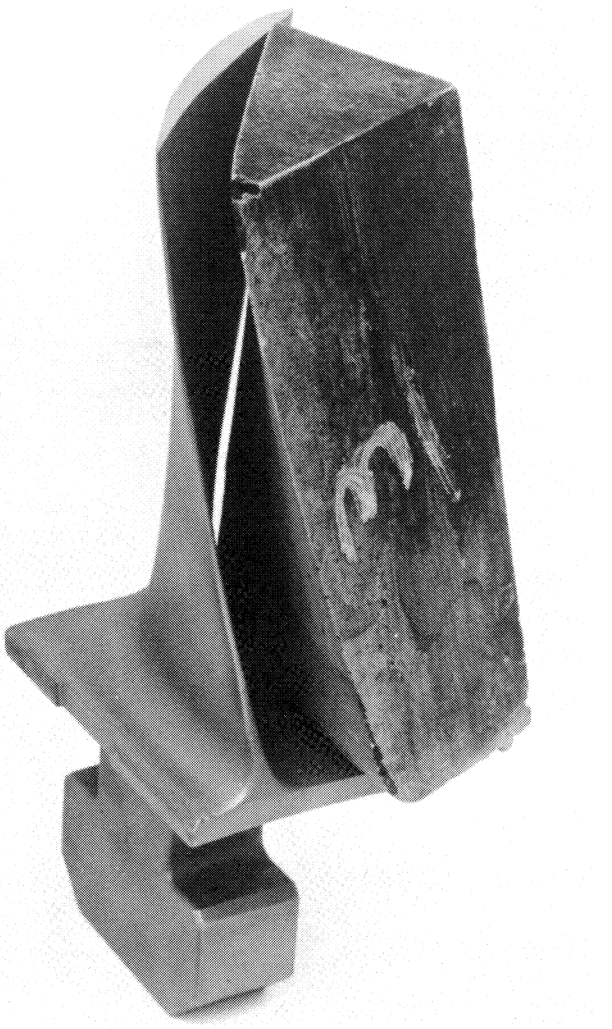

Figure 9. Template and Repaired Blade.

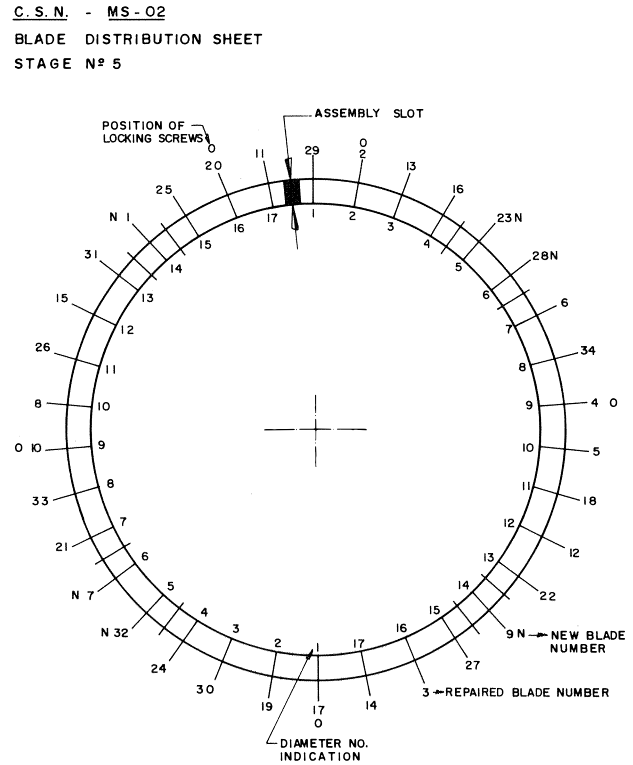

Figure 10. Example of a Weight Distribution Sheet for Rotor Blades.

In order to improve the resistance of the roots of repaired blades against fretting corrosion, it was decided to shot peen the lateral surface of the root. This process could inhibit potential microcrack development and crack propagation by fatigue. The surfaces of the blades were also micropeened with jets of glass beads oriented from the base to the tip to ensure an adequate and uniform surface finishing and reduce any residual surface stress concentration.

A total of 475 rotor blades were repaired and 107 new ones were used for assembly on the rotor to complete 11 stages. The blades of the first and second stages that could not be used, due to excessive airfoil damage were cut close to their roots. Those roots were used to cover the corresponding rotor grooves to prevent excessive turbulence. Broken roots were replaced by roots of the third stage unused blades. Balance weight holes were made on every other blade root of the first and second stages for a total of 17 holes in each row. Those two rows were intended to be used in conjunction with the existing balancing planes on both sides of the shaft in order to help in the balancing work if necessary.

Purchase of New Blades for the Sixth Stage

The sixth stage was particularly critical as its blades were the longest of the narrower profile ones, and they suffered the most severe damage. Besides, it had only six spare blades and obviously could not benefit any other blade cut to a shorter size. This stage alone could risk the expected performance of the blower or, worse, it could create surge problems by disturbing the pressure distribution and overloading other stages.

In order to solve this problem, a search was conducted for a known manufacturer in the United States that could supply new blades on an emergency basis. Forty new blades were purchased for this stage. Negotiations for a short delivery time of three weeks for such quantity of blades was possible because they had the necessary forgings in stock and modern fast production machines, in addition to fast reverse engineering resources based on a blade sample. The new blades were supplied strictly in accordance with the original sample dimensions, with superior finishing and with a shot peening treatment on blade roots.

Rotor Assembly

The blades were assembled on the grooves of the blower rotor and wedged so that the circumferential clearances found on the disassembly work were eliminated. Locking screws were used to fix them in their positions, and the first and last blades of each row were fixed by locking weld beads on both sides of the wedge. The wedge was then removed and the weld completed. All weldings were dye penetrant checked for cracks.

After finishing the assembly work, the rotor first and second stage rows were machined flush to the shaft surface to allow an undisturbed air flow. The tips of the other blades in each row were ground to their original heights to keep the previous existing clearances (Figure 11).

Stator Blades Repair

The repair work for stator blades was essentially the same as for the rotor blades, except that the damages were less extensive and there were many spare blades to replace the worst blades, as already mentioned. The drive shaft of each variable guide vane

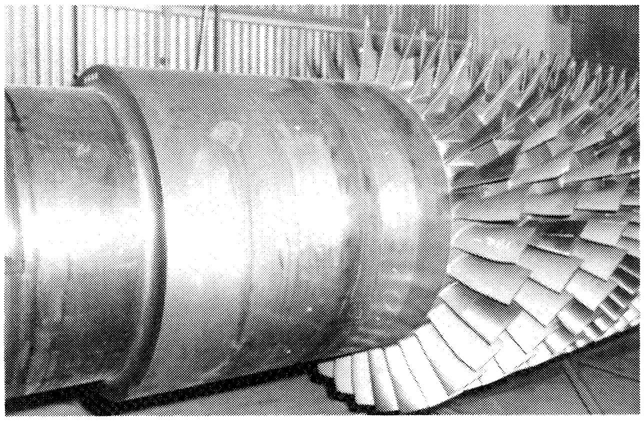

Figure 11. View of the Repaired Rotor.

was checked for straightness to ensure freedom of movement of rotation in their bushings, and each blade airfoil surface was checked for any bend or twist by means of specially made templates (see Figure 12).

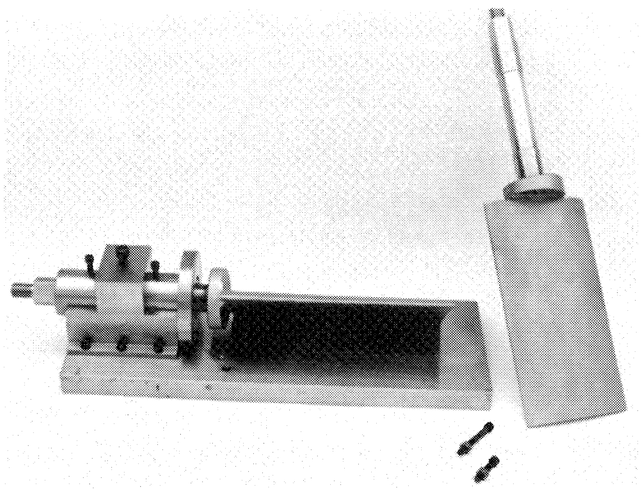

Figure 12. Template for Stem and Airfoil Straightness Check.

The inlet guide vanes (stage 0) and the first stage stator blades were discarded in view of the elimination of the corresponding rotor stages. A total of 274 stator blades were repaired, and 254 new ones were used for the casing.

New Inlet Guide Vanes

With the elimination of the original inlet guide vanes and the first stage stator blades, it became necessary to install a new set of inlet guide vanes at the inlet of the third stage. The time necessary to get special forgings, and the machining time to manufacture a complete set of new integral guide vanes, would result in a large increment of cost and repair time. Thus, a special design was developed in which the blade and its shaft were separately manufactured and silver brazed together (Figure 13).

The airfoil profile was simplified, but the leading and trailing edge angles were carefully studied to direct the air flow in an adequate way for the third stage rotor blades. The blades were made from AISI 420 steel plates and the shafts from 17-4 PH bars. The blades were cut from the plate so that their longitudinal axes were oriented in the same direction as the milling direction of the

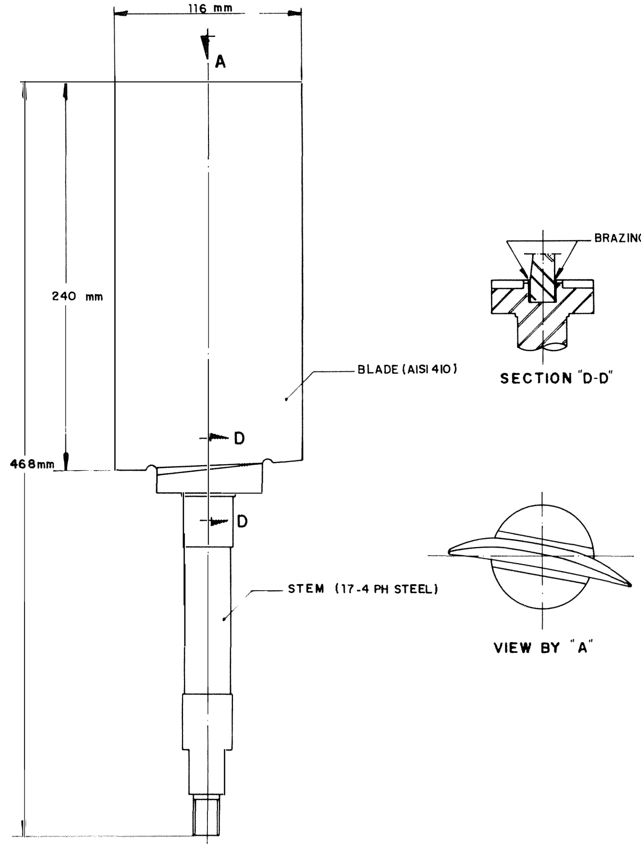

Figure 13. New Inlet Guide Vanes.

plate, and after being cut into shape and machined, they were subjected to a cold mechanical conformance to the right curvature. The shaft ends had a curved groove into which the blade root ends were fit. Brazing was made with AWS BAG-21 filler material.

Forty-three inlet guide vanes were manufactured, five of which were used in several tests to verify their strength. The inlet guide vanes were first subjected to ultrasonic and dye penetrant tests to ensure there were no internal flaws. After brazing the parts, a destructive test was made to ascertain that a minimum shear strength of 450 N/m2 was obtained. Two complete blades were subjected to impact and vibration tests to make sure that the brazed region would not fail. All test results were good. There were no failures.

Miscellaneous Parts

All stator blade slider pin assemblies which were damaged were manufactured in our shops. Special fixing washers and nuts for the blade shafts were obtained locally. DU guide bushings (special sintered material) for the stator blades were imported from France in view of the short delivery time. Damaged sleeves for stator blade shafts were replaced by existing spares and complemented by sleeves manufactured by CSN, which also repaired the slide plates for the control cylinder cage of the stator blade angle.

Casing Rework

In the plant, the cast iron stator casing was cleaned to remove all residual blade material that had been stuck to its internal surface due to the heavy friction. Nine holes were drilled at the bottom of the lower half (Figure 14) to install pressure gauges to monitor the pressures between stages as a reference for stator blade angle adjustment in case of surge problems. All burrs and nicks were

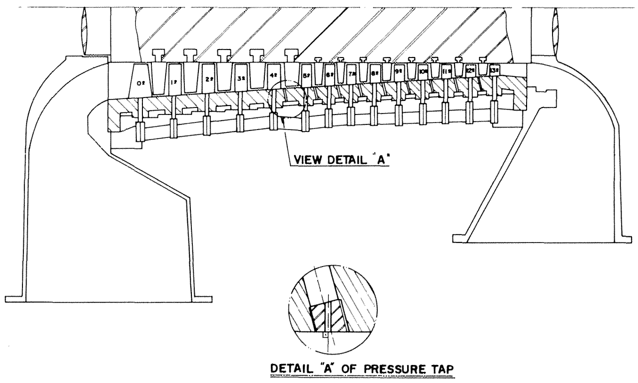

Figure 14. Lower Casing with Holes Made for Pressure Taps.

removed from the casing, and sharp edges were blended to avoid flow disturbances and also any stress concentration points.

DYNAMIC BALANCING OF THE ROTOR

In order to check the residual unbalance of the repaired rotor and rebalance it if necessary, the decision was made to use an overspeed chamber of a local manufacturer of large electrical machines. The rotor was assembled on custom made sleeve bearings supported by pedestals, in much the same way as any large generator rotor, and was coupled to the 1500 kW drive motor. It was expected that the rotor could be spun at the normal speed of 3600 rpm, despite the high expected ventilation loss due to the blades.

The speed could not go beyond 3300 rpm because of a power limitation of the drive system. An overhang critical near to that speed due to the unrestrained coupling end made the vibration increase excessively as the speed changed very slowly at the limit of drive power. The heating of the rotor due to the high ventilation loss made the air temperature in the chamber reach about 75°C during a short run, and the heat sensitivity of the rotor made it somewhat difficult to obtain consistent vibration readings. Nevertheless, several conclusions were obtained from the test runs that were made:

- The rotor was not heavily unbalanced, as the vibration crossing critical speeds were reasonable even without any correction weight, below 100 micrometers (3.94 mils) peak-to-peak. This was probably due to the uniform weight distribution of the blades, as previously mentioned, and the fact that the rotor did not show signs of a permanent bow.

- Polar plots during speedup indicated several internal loops. Some rubs with bearing seals occurred, and they were corrected, but the loops persisted in different forms. Attributed as the cause was the nonuniform heating of the rotor by the blades due to the different clearances between root and groove in new and repaired blades. The blades heated first with contact with air and transmitted the heat to the rotor surface by conduction during the speedup transient.

- During the cooldown of the rotor, it was noted that despite the tight assembly of the blades in the rotor grooves, they had become loose, and the impact between them was audible as the rotor was slowing down. This happened because the "fin effect" of the blades made them cool faster than the massive rotor, and the resulting contraction loosened them.

- The overhung effect of the coupling flange indeed had a large influence. A small weight on the flange (88 grams) made a dramatic change in the vibration amplitude. The same weight in any of the balance planes had practically no effect.

- The best compromise obtained in the dynamic behavior during runup was with 740 grams at the coupled end and 420 grams at the uncoupled end only four degrees apart. This result indicated that there was a reasonable static type unbalance compared to the rotor weight (about 2 times G 2.5 based on ISO 1940 Standard).

Based on such observations, and also in the experience had with the behavior of the Number 1 motor-blower during startup, it was assumed that the behavior of the rotor when coupled to the heavy flange of its drive motor should be completely different at high speed and most probably satisfactory. Due to the urgency to put the blower back to operation, a low speed balance check in a low speed balance machine in another facility was not attempted. The rotor was shipped right away to the field. All of this work took three full days.

FIELD ASSEMBLY WORK

The stator blades were assembled, except for the first and second rows, and the corresponding holes were closed by specially made plugs to eliminate the possibility of turbulence (whistling). The angular positions of the inlet guide vanes were checked and found to be satisfactory. The rotor was lowered into position, and alignment and clearance checks were made. The same original bearings were used after some minor cleanup work. The same discharge side labyrinth was used, but with all strips replaced. Two x, y proximity probes were installed at 45 degrees from the vertical centerline in each bearing case, in order to monitor the shaft vibration during startup and normal operation. A key phaser was installed near the coupling, and all probes were connected to their proximitors.

After closing the compressor casing, the driving system of the stator blades were reinstalled and all control systems were checked for proper operation. The pressure taps on the bottom casing were connected to pressure gages arranged in a rack besides the blower.

FIELD TESTS AND RETURN TO OPERATION

Trim Balance

The preparation work for startup included the setup of an instrument designed for data collection from the proximity probes installed on the machine to permit a proper analysis of transient data and a real time analyzer for prompt evaluation of dynamic conditions allowing immediate decisions about the corrective measures to be taken in case of any unexpected problem.

The first startup of the motor-blower was made on November 25, 1991. The speed was first increased up to 800 rpm and then cut down to check for any rubs or abnormal noises. Everything was normal, so the blower was started up again, and its speed increased to 3,600 rpm with no apparent problems. The motor was synchronized and the train was left running to stabilize the operating conditions. The polar plot corresponding to the highest shaft vibration, which was on the coupled end horizontal position, is shown in Figure 15. The levels were satisfactory, as the higher amplitudes crossing criticals were below 80 micrometers (3.15 mils) peak-to-peak filtered at running speed and reached a maximum value of 80 micrometers at the coupled end horizontal position at full speed. However, the amplitudes were slowly increasing with time in a typical "thermal effect" fashion (see Figure 16), and it was decided to attempt a refinement to bring down those levels. The trim balance took some time (two days) due to the need to wait for the machine to cool down to install or remove weights for each trial run.

Eventually installed were the same weights defined at the overspeed chamber, both rotated 30 degrees against rotation compared to their original position. The final vibration readings are presented in Figures 17 and 18 for the same coupled end horizontal

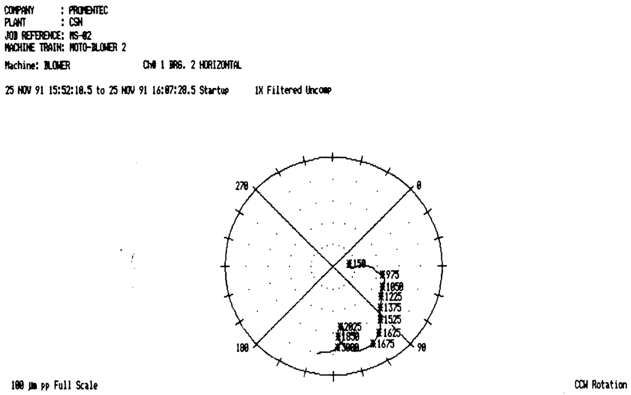

Figure 15. Polar Plot During First Startup after Repair.

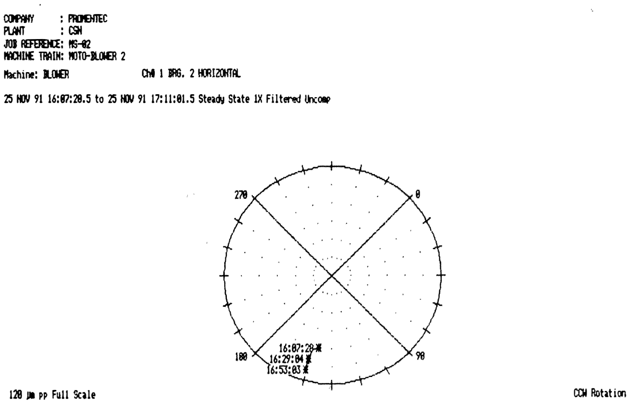

Figure 16. Polar Plot Showing Vibration Change with Temperature at Constant Speed.

direction, which still had the highest filtered amplitudes. The overall amplitudes compensated for runout were below 60 micrometers (2.36 mils) peak-to-peak. The axial casing vibration was somewhat higher than desirable (maximum of about 7 mm/s or 0.275 mils overall peak velocity), probably due to a residual misalignment or axial aerodynamic excitation, but that was considered acceptable, provided it did not increase with time.

Performance Evaluation

The performance limits of the blower were checked and found to be superior than previously announced to the company. The

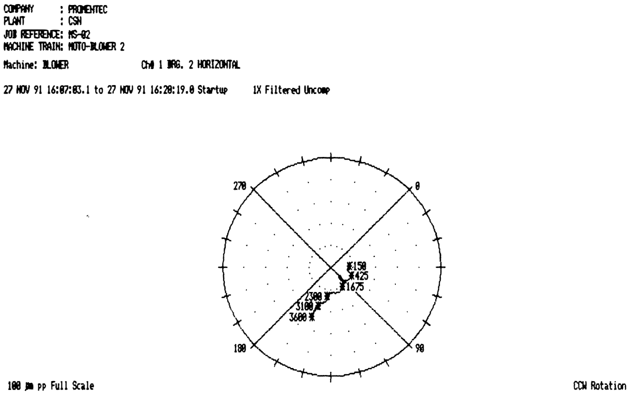

Figure 17. Polar Plot of Startup after Balance Correction.

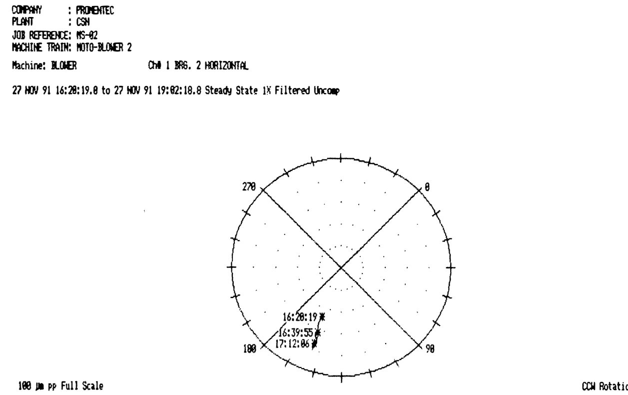

Figure 18. Polar Plot Showing Vibration Change with Temperature At Constant Speed after Balance Correction.

several performance points that were obtained are indicated in Figure 19. In particular, the point 4 was enough for the blower to supply air to the blast furnace Number 3. The previous expected condition was intended to supply air only to the blast furnaces 1 and 2, which required less air flow.

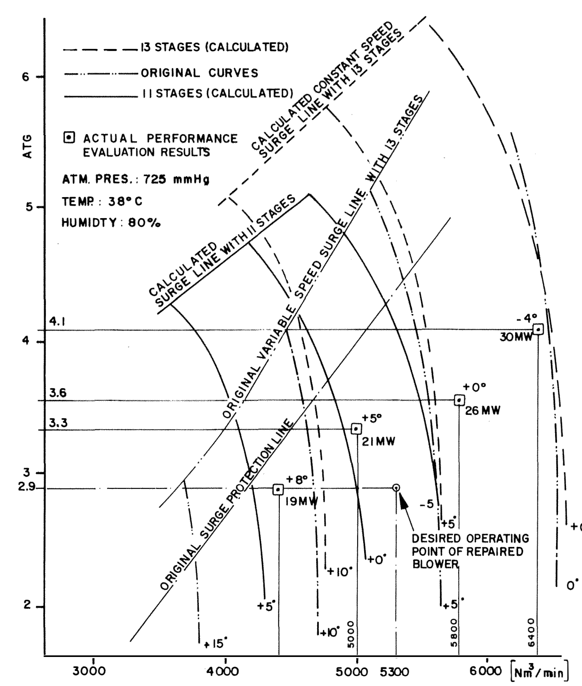

Figure 19. Results of Performance Evaluation.

The company considered the test results very satisfactory, and they demonstrated complete satisfaction in their decision to make the temporary repair in spite of all manufacturer recommendations against it.

THE ACCIDENT WITH THE NUMBER 1 BLOWER

While the damaged Number 2 motor-blower was out of operation with the case opened, and the company was looking for a suitable solution, the Number 1 motor-blower was shut down on July 30, 1991, due to a sudden increase in the vibration levels at the bearing casings. At first, company personnel investigated a possible bearing problem, since there was a change in the oil pressure of the uncoupled side simultaneously with the increase in the vibration. At the same time, the consultants were called to the site to make a vibration analysis during startup. However, during the startup after bearing check, a clear internal noise of metal-to-metal contact could be heard at 500 rpm. The blower was then opened, and it was found that one of the 13th stage rotor blades had ruptured and affected all blades of that stage, as well as the stator blades of the 12th and 13th stages.

In analyzing the situation, it was decided to remove all blades of the 13th rotor stage and to put the blower back into operation, because the Number 3 blast furnace was loaded and could not stand a long period without air. Thus, there was a real risk of complete loss, since there was no blower available to feed it with air.

It was expected that by removing all rotor blades of that stage the residual unbalance would not be adversely affected as the radial symmetry would be kept. The trailing edges of the 12th stage stator blades were bent near the tips and attempts were made to correct them, although the results were not completely satisfactory, due to the short time available.

The plant maintenance personnel did a remarkable job as the blower was opened and the 13th stage rotor blades removed and assembled back in only 36 hours. The blower was started up on August 4, 1991, with extreme care, and the work proved successful as the vibration level was very good. The blower is still in normal operation and it will be shut down for a complete inspection of the blades as soon as possible.

The cause of the rotor blade failure was investigated. One of the 12th stage stator blade rotational arms was not engaged in the control ring, probably during assembly. Thus, the most probable cause of this failure was the resulting heavy alternating load on the rotor 13th stage blades, and the weakest blade first developed a fatigue crack. No cracks were found on other blades of that stage on dye penetrant tests.

CONCLUSIONS

The first conclusion from the good results obtained with the Number 2 motor-blower is that a temporary repair work, even in a complex case as presented, can be successfully done and can drastically reduce the time necessary to reestablish production and avoid large revenue loss. The second important conclusion is that the work, when properly planned and conducted, permits it to maintain the schedule and can guarantee successful results. The original estimate of three to four months proved to be correct as the time spent for the repair was 92 days (Figure 20), while the total outage period since the accident of the motor-blower was 165 days. The third conclusion derived from the incident with the Number 1 motor-blower is with such equipment and such maintenance, things could have gone even worse than they did, but a creative emergency solution such as the one implemented, which has been described in this paper, finally contributed to minimize company losses.

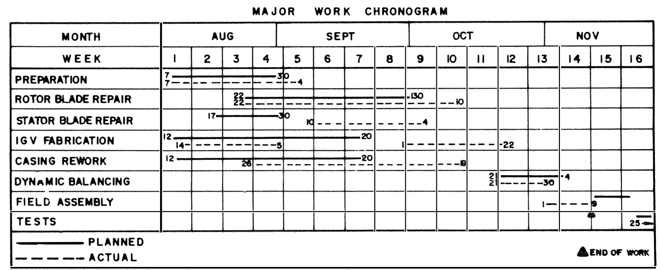

Figure 20. Comparative Planned vs *Real Chronogram.*

As a final comment, it would be good to point out that proper maintenance of these blowers should necessarily contemplate a careful inspection of the blade roots with respect to fatigue cracks. The manufacturer's maintenance schedule did not take into consideration the need for such an inspection. On the other hand, a shaft vibration monitoring system could eventually permit the detection of the development of cracks in the blades and allow corrective measures to be taken, or at least trip the machine by excessive vibration before it reaches a level that could cause machine destruction.

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