COMPRESSOR AND TURBINE SHAFT REPAIR
A USER’S VIEWPOINT

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ABSTRACT

Very stringent reliability requirements for large unserved compressors and turbines used in refineries and chemical plants demand the use of considerable judgment when repairing and salvaging shafts. Most repair techniques; flame and plasma spray coatings, detonation gun applied coatings, chrome plating, nickel plating, welding, weld overlaying, cutting, sleeving, and new shafting; are used to make the repair urgent, plant reliability, and machine reliability compatible. The objective is to review machine reliability, to classify repair methods, and to help the user select a good sound repair package.

INTRODUCTION

Compressors and turbines used in a typical refinery are usually unserved machines with drivers from 1,000 to 12,000 horsepower. Some of the newer chemical plants have unserved machines with 25,000 to 90,000 horsepower. These machines are expected to run one to five years without any shutdown for maintenance. These long runs keep operating and maintenance costs low; however, the anxiety about a reliable repair job is rather high. Completely riskless repair decisions are not always possible.

Compressor and turbine repair as well as shaft repair decision-making should consider time, machine reliability expected, tangible resources available, such as materials, spare parts, shop and field tools, cleaning, grit blasting, welding, coating, plating and heat treatment as well as some of the less tangible resources such as numbers of people available, inventiveness of the people, and craftsmanship. Can the user afford to make a less reliable but not unacceptable repair, a more reliable repair or a maintain-position repair relative to equipment manufacturer’s quality while an oil or chemical product producing plant is idled? Can the user afford to increase the repair time of a shaft or rotating element in order to upgrade it and risk failure of the active machine in the operating plant? A look at the reliability of various machines and a look at the various repair options will help guide the user’s judgment.

RELIABILITY OF MACHINES

Figure 1 shows a pump reliability curve for a refinery with a few chemical processes. The sample used for this curve was large enough that it should establish a curve shape for comparison while examining the smaller populations of larger turbines and compressors. A number of these curves have been made for other types of machines and there are invariably two median maintenance repair intervals. The curve for pumps shows a one-year median life for about 10% of the population and a 10-year median repair life for another group, 90% of the population. Examination of Figure 2 shows a group of centrifugal compressors, 15%, with a median life of two years and another larger group, 85%, with a median repair life of five years. Another observation is that the 13, 16, and 18-year compressors may have economically required overhaul at about a 10-year life. The two and three-year life group requires prompt action on spare parts and rotor repair. Extra money can probably be spent to upgrade the machines and the chances for improvement are great. The upgrading efforts are difficult to arrange because the repair should be done in about four to five months to avoid being caught by a surprise failure of the operating machine. The machines with an overhaul life of five years or more can be allowed one year for repair without great risk. Experience would say that the user would get caught about once in 20 years if 20% of the repair interval was used as an acceptable time for spare rotor repair.

Figure 3 is a reliability curve for large steam turbines. The population is small and the reliability is high, which makes the

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Figure 1. Pump Reliability for a Large Refinery With a Few Chemical Plants.
curve incomplete but the shape is there — two median reliabilities, one at four years and one at about ten years. Again, the twenty-two-year, seventeen year, and fifteen year machines are probably beyond the time when fuel economics would say that they need an overhaul. Twelve or thirteen years look like a reasonable maintenance interval. The same comments made about compressor rotor overhaul time would apply to turbine rotor overhaul.

An important observation concerning turbine and compressor overhaul is that about one-third of the shafts and rotating elements are reassembled with the machine when it is overhauled. New labyrinths and other wear parts are replaced and minor repairs that would not increase the anticipated shutdown interval are made to the shaft and rotating element. About two-thirds of the shafts and rotating elements require repairs that would extend the overhaul by a week to a month and occasionally three months if spare rotors were not available.

Establishing sources for emergency work; rotor stacking, balancing, bearings, seals, labyrinths, stationary parts castings, and machine work, is also an important part of successful compressor and turbine maintenance.

**CHROME PLATING REPAIRS**

Figure 4 shows a typical compressor rotor used in the refinery. Chrome plating has been successfully used to remove a gentle bow of several thousandths of an inch from a shaft by plating and regrinding it concentrically with the journals.

Plating repairs have also been made to a surface fatigue-failed shaft under an interference-fitted thrust disc; the most troublesome thrust discs are those that are one inch to one and one-half inches thick in the axial direction. This repair is probably ten thousandths of an inch thick or more and must be machined smooth before plating and must be reground to size after plating. The repair is usually permanent. Thin thrust discs appear to wobble or vibrate on the shaft lifting thin flakes of metal from the shaft. Most large horsepower compressors (3,000 horsepower or more per casing), where shaft surface fatigue is a problem, have enough aerodynamically excited vibration at the natural frequencies of thrust discs to excite their vibration. This idea has been examined and given up as difficult to prove; however, it is a fact that two to two and one-half inch thick thrust discs or discs with a substantial hub almost never surface fatigue the shaft. With thick discs, there is a slight etching of the shaft that takes many years to become significant enough to require repair (See Figures 5 and 6).

The tapered coupling end of compressor or turbine shafts can be permanently repaired by chrome plating. This damage is very similar to the thrust disc-caused failure. Very minor overloaded high spots under the coupling hub fatigue and lift out of the shaft in thin flakes. If the coupling has been lapped to the shaft, the parts that are well mated will have no fatigue but when the coupling hub is installed by forcing it up the shaft taper twenty-five to fifty thousandths of an inch onto an unplanned portion of the shaft (a shoulder), a ring of fatigue failures will occur where the large end of the coupling bore overloads the shaft surface. This concentrated ring of shaft fatigue has caused numerous shafts to break especially when combined with the shaft bending moments induced by a worn or "locked up" coupling. This surface fatigue-failed ring at the top end of the shaft taper can be avoided by lapping to a master ring and plug or by removing a little metal from the ID of the coupling hub at the large end after lapping. If lapping has not been overdone, the material can be removed from the coupling hub bore by hand with a scraper.
Chrome is a strong material and if deposited thickly enough, the shrinkage will cause cracking of the plated part. Some platers attempt to eliminate the potential for cracks by applying plating in thin layers, shot peening each layer before proceeding. Cracking of chrome plating can be induced by applying part of the coating at a different current density from the rest causing one layer to be much stronger than the other. Chrome plating is also very porous and it will be popped off by rusting of the backup material if applied in a corrosive service. An undercoating of nickel plating will sometimes solve this problem. It would probably pay to be very familiar with your plating shop and the capability of its people.

**FLAME SPRAYED COATINGS**

Flame sprayed coatings are successfully used in great quantities for pump shafts, low speed gear shafts, mechanical seal sleeves, small turbine journals, carbon ring surfaces, and surfaces requiring abrasion resistance without hard rubbing or localized heating. In contrast, some of the coatings are fused to a blank during manufacture of a new part such as a pump wear ring or a packing sleeve by heating the entire part to a bright red, wrapping it with insulation and laying it aside to cool slowly. The coating is then resistant to hard rubbing if the backup material is moderately hard (25-45 Rockwell C). A great deal of experience with shrinkage and distortion is required to fuse a coating to a blank of the correct dimensions so that the hard surfacing will not show up in the bore of the nearly finished part and make it nearly impossible to machine. Most repair shops will not make parts with fused coatings that have to be machined to specific dimensions at a reasonable price.

Flame sprayed coatings have been used with limited success on some of the large unspurred turbines and compressors where high reliability is required. In desperate cases, journals, balance pistons, shaft sleeves, shaft under thrust collar and tapers for couplings have been coated, but life of the parts has been an unpredictable one-to-four years. Aluminum compressor labyrinths and thin ribbeted bronze bearings can be made in the users own shop or obtained from specialty machine shops in a very short time which makes it very easy to cut down damaged shaft areas and fit new stationary parts. Shafts are not cut down without at least a rough machining of stress levels and the shaft natural frequencies. If enough stress calculations and shaft natural frequency tests are made, the feasibility of a shaft modification is easy to evaluate quickly.

Four condensing steam turbines (2,500 hp, 5,250 revolutions per minute, 180 pounds per square inch steam inlet) and one back-pressure turbine (6,000 hp, 7,200 revolutions per minute, 650 pounds per square inch, 750 degree Fahrenheit steam inlet, 180 pounds per square inch, 500 degree Fahrenheit exhaust) are fitted with carbon rings running on flame spray-coated surfaces. They are not very reliable and they account for the hill population at four years on the Figure 3 steam turbine reliability curve. The reliability is four years because the plant runs four years. The life of the coatings has been unpredictable. Sometimes a four-year run is made without leaks, however, some terrible leaks have been tolerated to keep the plant onstream. Two years ago the one-eighth inch monel coating was removed from the back pressure turbine shaft and it was replaced with a thirty-thousandth inch thick coating, No. 442 (73% nickel), which was machined and polished but not ground (See Figure 7). A diaphragm leak inside the machine was corrected to reduce the pressure sealed by the carbon rings and special oversized carbon rings were made to fit the shaft. This combination seems like it would last longer, but probably not for more than four years. The spare rotor for
TURBINE 6000 HP
FLAME SPRAY AND DETONATION
GUNNED COATINGS

Figure 7. Back Pressure Turbine Has Failed Coating Under Carbon Rings and Labyrinth Groove Failure.

the back pressure turbine was also cut to a smaller diameter in the carbon ring area to remove the monel coating and it was coated with detonation gun applied tungsten carbide. A gland condenser, a set of labyrinths to replace the carbon rings and a set of undersized carbon rings are held for use with the spare rotor when maintenance is required. The goal is to replace the carbon rings with labyrinths if repair time permits.

The four 2,500-hp condensing turbines have to have the entire casing removed for replacement of carbon rings on the balance piston and two shaft packing areas. The shaft repair to the working surfaces could be reduced to insignificance by use of detonation gun applied tungsten carbide. The carbon rings would then be more reliable; however, it is doubtful that they would last a second run which would be eight years. The amount of work to replace carbon rings by removing the turbine casing is still very significant.

SLEEVING REPAIRS

Sleeving has been proposed for turbine and compressor shaft repair numerous times but it always seemed like an unnecessary risk. The thought was that a bearing failure, even though moderate, would cause removal of the rotor from the casing. The sleeve would have been interference fitted over the journal to return it to standard size. Even though sleeves were considered a risk, compressor manufacturers protect a shaft in the floating ring seal area with comonoy coated sleeves and the wear areas between impellers are protected by sleeves that also serve as impeller spacers. A small space is left between the impeller and spacer to prevent shaft bowing during mechanical tightening of the retaining nuts or during operating thermal transients. The space left is small enough to protect the stationary parts from being rubbed by an impeller and to avoid loss of performance if an interference fitted impeller should move. Both of these sleeved area control the overall life of many compressors. Detonation gurned tungsten carbide is expected to improve the life of these sleeves for at least two years.

DETONATION GUN APPLIED COATINGS

Detonation gun applied coatings are the most durable coatings used for the repair of large unspared compressor and turbine wear parts and shafts. The coating, if applied to outside diameter of parts, adheres (greater than 25,000 pounds per square inch) well enough to enhance the reliability of a machine. In contrast, many laboratory-applied, flame sprayed coatings adhere in the range of 3,000 to 10,000 pounds per square inch. Many machinery parts with detonation gurned coatings subjected to really severe services have lasted for over ten years without a single wear-out. Compressors and turbines require repair so infrequently that the use of detonation gun applied coatings has been limited to the less reliable machines which is exactly where it should help most. In compressors and turbines, the coatings have survived in all but the very hard rubbing cases where the metal behind the coating has yielded. One of the tungsten carbide coatings has been used for coating of compressor impeller spacers, balance, pistons and floating ring seal sleeves. Since the parts had to be sent to a coating shop for repair, the shaft was coated on the coupling taper, journals and thrust disc fit as required.

One outstanding case was a single stage, 6,000 revolutions per minute, 1,000 horsepower condensing steam turbine driven blower for a sulfuric acid plant. This compressor handles "dry" sulfur dioxide which should not be very corrosive. The sulfur dioxide leaked from the compressor labyrinth packing into the tube oil system where it combined with steam condensate from the turbine carbon ring packing, causing journal corrosion approaching fifty to sixty thousands of an inch per year. An entirely new compressor shaft, including the integral thrust disc and all working surfaces of a new turbine shaft (journals, carbon ring and bearing seal surfaces), were coated with a selected detonation gun applied tungsten carbide coating which raised the blower and turbine reliabilities from one to three to five years. A dry air purged carbon ring sealing system was installed on the compressor which has a much lower leakage than the original labyrinth seal; however, the compressor and turbine carbon rings have to be replaced annually, but neither machine casing has to be opened to do this job. The general reliability of the whole acid plant is so much lower than an oil processing plant that this turbine and compressor performance looks good.

There are several comments concerning best use of these coatings:

1. Do not apply a detonation gurned coating over a flame or plasma sprayed coating because the adherence will be reduced to that of the undercoating. The coating is expensive and should not be downgraded.

2. Try not to apply a detonation gurned coating over a backup material less than 30 to 45 Rockwell C hardness. The backup material will yield and the coating will break-up. Some compressor parts are held to 25 Rockwell C to prevent hydrogen cracking. Contrary to the appearance of the uncoated parts. they must be abraded rather than yielded in hard rubbing. Detonation gurned coatings have been successful on these parts.

3. Do not apply detonation gurned coatings to an inside diameter. The applicators claims appear to be in good faith and performance is improving but experience so far is poor.

4. Use GA mehanite or Ni-resist 2B for parts requiring bores to run against a detonation gurned coating. Both are low friction materials (.06 and .1 friction coefficient in dry rubbing) and Ni-resist 2B is a very low wear material.

5. Do not apply the coating more than ten thousandths of an inch thick.

6. Aluminum labyrinths seem to work well against detonation gurned, tungsten carbide coatings even though...
they make a bed for abrasive particles from the process.

NEW SHAFTS

Shaft repair needs to be weighed carefully against the time and material costs for making a new shaft. If a shaft is upgraded by welding, coating, plating and grinding, then it may be worthwhile to spend more time on it than would be required to make a new shaft. If a new shaft is made, the quality of the material is of concern. Some manufacturers of high speed, high quality machines buy special forgings for shafts. The refinery makes any shaft up to four inches in diameter and a few are made from six inches to twelve inches in diameter. Rolled bar stock (usually AISI 4140 or 4150) is used for almost all shafts with good success. The machinists who make shafts have a difficult time remembering a material defect, one in twenty years seems to be typical. The risk of defect increases as the size of the bar stock or forging increases and should be part of the decision-making process.

REPAIR BY STRESS RELIEF AND SIZE REDUCTION

Figure 8 shows the rotor of a 7,000-hp, 8,000 revolutions per minute, six-stage, extraction-admission turbine which has wheels machined integral with the shaft. The speed governor of this turbine failed after about one year of service which caused a severe coupling end bearing failure when the speed climbed to 11,000 revolutions per minute, the overspeed trip setting. The vibration made the bearing housing egg shaped, the journal was heavily scored, the blue-black journal was hardened to about 50 Rockwell C and the shaft was bent about one-fourth inch. A normal hurried repair probably would have been to turn the shaft and journal down about one-half inch, and put the machine back together with a modified bearing and coupling. However, a spare rotor was installed in the machine and a repair was sought that would correct the vibration problem as well as the bent shaft. The manufacturer recommended a new rotor because he feared fatigue at the stress concentration caused by the hardened interface near the journal. A new rotor would have cost $250,000 in 1972. The hardened journal was unsatisfactory to the refinery because hydrogen sulfide, chlorides and moisture were expected to be present in the oil reservoir at various times. The corrosive environment caused by chlorides and sulfides would reduce the fatigue life to one-third of the calculated value and hydrogen sulfide causes cracking of materials harder than 22-25 Rockwell C. Stress relief of this journal with a pipe stress relieving machine was mutually accepted after much discussion between the mechanical engineers and metallurgists. A temperature of 1,050 degrees Fahrenheit and a time of four hours were chosen which would draw the hardness of the journal down to about 25 Rockwell C without softening and thus weakening the adjacent metal. The first attempt failed because the temperature was too conservatively chosen and because the large shaft and wheels conducted more heat away than anticipated. The manufacturer suspected the trouble, found some pieces of the shaft material, abused them to create samples with hardened spots and did a laboratory stress relief on the samples to find a correct, less conservative temperature and time. A new temperature of 1,080°F and a time of 16 hours was tried on a thoroughly insulated rotor. The second stress relief was successful leaving all parts about 25 Rockwell C. The shaft was then turned down about one-half inch and shortened two inches to complete the repair (cost $33,000). Tests were conducted on the turbine containing the newly installed rotor and two problems were found: (1) a second shaft natural frequency at about 10,000 revolutions per minute and a subsynchronous vibration that appeared about 8,700 to 9,000 revolutions per minute which excited the first natural frequency of the rotor at 6,000 revolutions per minute. Another subsynchronous vibration appeared at 3,000 revolutions per minute. This system would start to vibrate noticeably at about 8,700 revolutions per minute and peak at about 9,000 revolutions per minute but the vibration magnitude would not decrease with decreased speed until the speed was lowered below 6,000 revolutions per minute, the first shaft natural frequency. The temporary cure was to cut away about one-third of the babbitt in the lower half of the coupling end bearing (a strip one and one-half inches wide was cut from the center of the bearing leaving one and one-half inch wide strips at each end) to put extra load on the oil film and to install a light-weight coupling (See Figure 9). The final solution was a tilting shoe radial bearing and a very light-weight coupling on the shortened shaft. The second critical speed was raised to 15,900 revolutions per minute by "hump test" and the subsynchronous vibration has made no further appearances.
Since 1972, three high-low labyrinth failures have occurred: the exhaust end labyrinth packing, the balance piston labyrinth, and the high pressure-end labyrinth packing. The exhaust end labyrinth failure shown in Figure 10 did no damage to the rotor; however, the grooves in the balance piston area and the high pressure packing area were cracked, "heat checked," enough that the grooved areas had to be remachined and smaller bore labyrinths purchased as shown in Figure 11. The labyrinth tooth location in each case had been moved about 1/16 inch toward the exhaust end to accommodate more shaft expansion. Errors in setting the thrust bearing and/or nozzle clearance could have caused this problem. From the operating side, overheating of the rotor by running without vacuum on the exhaust or running the condensing end of the turbine on low pressure steam without sufficient cooling steam in the high pressure end could cause heat from windage to expand the shaft enough to cut off labyrinths.

Figures 10 and 11. Turbine Labyrinth Failures.

OTHER REPAIR METHODS

Rather than leave the thought that conservative compressor and turbine shaft repair practices are the party line for all shaft and equipment repair, several illustrative cases should be described.

Low-Speed Gear Shaft Welding

Rather severe tooth wear sometimes appears in low speed gears. The problem is not without warning because the gears commence to "wow" long before wear becomes significant, but diagnosis is difficult and finding the right combination of people willing to spend an inordinate amount of time to find and correct the problem is also difficult. Misalignment, or a worn locked-up coupling, is a frequent cause for tooth wear in sleeve bearing gear units. When the teeth are worn enough to warrant the effort and when replacement gears are not at hand, the gear and pinion can be turned over to use the other side of the teeth as shown in Figure 12. The shaft is bored out of the gear, the gear is turned over and shrink fitted to a new shaft. Unfortunately, the pinion shaft is integral with the toothed section in most cases; therefore, a reference area is turned on the pinion shaft very close to the toothed section, the coupling end of the pinion shaft is cut off just outside the journal, the pinion is turned over and a new section of shaft is welded to the end of the old shaft so that a new journal and coupling taper can be turned on it. The reference areas are used to set the pinion up in a lathe and later in a grinder to be very certain the pitch line is made concentric with the shaft centerline. Low-speed gears (1,800 rpm to 400 rpm) have lasted 3-10 years after this treatment. The life depends on the care used in turning and grinding the new pinion shaft. Welding of an AISI 4140 HT shaft without stress relief is metallurgically unacceptable due to the hard interface of the weld heat affected zone, but it seems to work. Corrosion is not usually present to accelerate fatigue in a gear box which may help account for the success. Welding a fatigued-failed shaft together should be considered carefully because the stress required to cause a fatigue failure is already present and the hardened interface at the weld will cause an accelerated failure.

 Shaft Overlay

An overhung propeller pump circulating 80,000 to 100,000 gallons per minute of hydrocarbon and concentrated sulfuric acid in a reactor had considerable shaft corrosion between the impeller and the mechanical seal. The AISI 4140 HT shaft was repaired by weld overlaying it with Alloy 20 which is resistant to sulfuric acid corrosion. The shaft broke at the hardened interface, heat-affected zone of the weld, immediately behind the impeller after about one month of service. The corrosive environment and hydraulic impact from cavitation no doubt accelerated the fatigue failure. A weld overlayed, unstress relieved, low carbon steel shaft also broke after one year of service (See Figure 13). There should have

Figure 12. Turn Gear Over; Weld on New Shaft, Use Other Side of Teeth.
been no hardened interface between low carbon steel and an Alloy 20 weld overlay; however, the low carbon steel is only about one-half as strong as AISI 4140 HT. No plain AISI 4140 HT shaft without overlay had ever broken in service and the corrosion of the shaft had never caused a reactor shutdown. In this case, a new shaft appeared to be the best repair.

Electric Motor Shaft Welding

Electric motor shafts are frequently destroyed at the journals when a bearing fails. A flame sprayed repair would be used if the deposit required was not over one-sixteenth of an inch thick. For more serious damage, the laminations and squired cage can be pressed off the shaft and installed on a new shaft, but the machinists seem to prefer to cut the damaged shaft off near the fan and to weld a new piece of shaft material to the old shaft that can be turned to the original dimensions (See Figure 14). This again is a metallurgically unacceptable repair due to the exceedingly hard interface at the weld of AISI 4140 HT to AISI 1020. The repair method has worked; no shafts have broken in service.

Pumps Shaft Coating

Pump shafts have been successfully coated with zirconium oxide, ten to thirty thousandths of an inch thick, to slow the heat input of an impeller to the shaft in a hot, 600 degree Fahrenheit pump. Reducing the heat transfer to the shaft cooled the shaft and bearings enough to allow removal of a very undesirable bearing waterjet that was causing bearing failures by over-cooling the ball-bearing outer races, which caused the bearing to seize. A coating of zirconium oxide has been added to the mechanical seal sleeve to lower temperatures in the mechanical seal environment, the stuffing box. Another coating of zirconium oxide has been added to the hot process side of the stuffing box, inside of the head, to reduce the heat transfer from the process liquid to the stuffing box water jacket. This coating has successfully increased waterjet fouling time from a month or two to a year or two. Zirconium oxide is not a tough coating and it will need some repair each time a pump is repaired, however, this is acceptable if the pump life is increased and if the flame spray coating can be repaired quickly (See Figure 15). Temperatures of identical pumps in pump and spare services were compared, one with zirconium oxide insulation and one without. A 100 degree Fahrenheit reduction in shaft temperature at the face of the mechanical seal flange can be expected when a 600 degree Fahrenheit process liquid is handled. A carefully made 70 percent hollow shaft was installed in a large overhung pump to improve the mechanical seal and bearing environment. This experiment should have reduced the heat transfer by 70 percent. The results appeared to be good but tests were not possible due to the critical service of the pump. A hollow shaft is a more durable solution to the bearing-seal environment problem.

Shaft Slewing

A large boiler fan failed a bearing and scored the shaft severely. Other equipment had been installed around the fan so that the casing could not be removed by unbolting it in a conventional manner. The casing would have to be torch-cut in half and welded together again after the fan rotor was reinstalled. The fan was large enough that handling it in the machine shop would have been very awkward. The failed journal was dressed to remove high spots. The shaft was indicated to be straight, so a sleeve was made and shrink fitted over the journal. The bore of the sleeve was fitted to undamaged areas on either side of the bearing journal. The bore of the sleeve was larger in the center so that it would not be distorted by fitting tightly over the irregularly scored areas of the journal. A larger pillow-block bearing was installed with a bearing to fit the new larger journal. Fortunately, there were enough shims under the old pillow-block bearing that the new bearing could be installed by shim and gout removal only. This fan has not had trouble for a number of years.
SHAFT REPAIR CONCLUSION

Shaft repair methods suitable for compressors, turbines and other machines, exist in rather large numbers and varying reliabilities. There are also varying opinions concerning the reliability of the methods, most of them related to welding procedures, thickness of coatings, materials, craftsmanship, and previous service of the part. Therefore, the user must form objective opinions about the usefulness of repair methods as they are related to reliability in his various refinery and chemical plant environments. The repair methods must also be classified relative to the plant reliability. The user who knows his needs must then accept his share of responsibility for the repair and guide his own repair crews or he must contract repair crews into suitable repair strategies that will restore original reliability, will last until the next plant maintenance overhaul, or will upgrade the part or machine depending on the situation at hand. Some user inspection may even be required to assure good craftsmanship and good communication.