DESIGN AND OPERATION REFINEMENT OF A MULTISTAGE LIGHT HYDROCARBON PUMP

by

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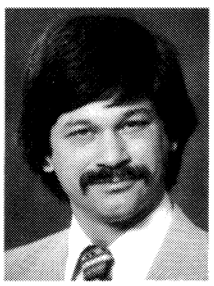

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Previously Mr. Laplant was a Staff Mechanical Engineer in the Maintenance Department of Oxy Petrochemicals, Inc., at the Chocolate Bayou petrochemical plant, where he provided technical support to all of the departments within the plant. For the last 13 years, he was heavily involved in troubleshooting, specification, repair, and upgrade of plant machinery. Mr. Laplant also took a special interest in optimizing pump and mechanical seal reliability at the Chocolate Bayou site. This involved implementation of modern techniques which resulted in improved equipment run times. He also has been involved in plant startups, turnarounds, and problem analysis while in the maintenance departments of three different plants.

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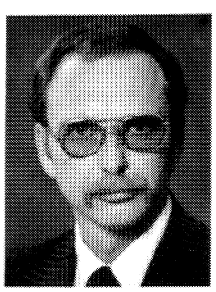

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Initially, he was a Project Engineer for Monsanto Company, then moved into equipment specification, installation, startup, and problem solving. After Monsanto, he worked for Conoco Chemicals, Dupont, and Cain Chemicals.

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ABSTRACT

One of the most critical pieces of equipment in an Olefin plant is the ethylene product pump. The challenge of designing a vertical process pump that will deliver a high vapor pressure liquid with poor lubricating qualities at 10,000 ft TDH and stay within budgetary constraints is understandably an enormous task. Often process and pump designers do not adequately appreciate the need for these pumps to have the ability to withstand periodic abuse resulting from plant upsets and human error. Also, purchase specifications for the design of the pump and pump system do not always coincide harmoniously with the actual system requirements. As a result, these pumps go through a never ending refinement process of design as the user company and pump company strive to improve the MTBF as process conditions change and technology evolves.

Experiences on such a system at one petrochemical facility can be categorized as follows:

- Installation of supply reservoir, piping and valves
- Operating philosophy and methodology
- Driver design and installation
- Reliable method of instrumentation for vibration monitoring and analysis
- Pump parts design for reliability and ease of maintenance
- Foolproof mechanical seal installation
- Coupling design and manufacture
- Driver to pump alignment

These topics are discussed from the practical viewpoint of Engineers practicing at the plant level in a Gulf Coast petrochemical facility.

INTRODUCTION

The equipment discussed herein consists of two installed vertical $4 \times 8 \times 10\frac{1}{2}$ double case pumps with a 20 stage opposed configuration. The pump, originally supplied with 20 impellers, is capable of producing 9,300 ft TDH at 550 gpm from a source of liquid ethylene at −37°F and 325 psig. The pump takes suction at the bottom, approximately 10 ft below grade, then pumps the liquid up through 10 stages where it passes through a cross over channel to the top of the pump. The flow then continues down through the final 10 stages to the center of the inner pump case where the liquid is channeled into the outer case. It then travels up to the discharge flange through the annulus formed by the inner and outer cases.

The pump is set in a 10 ft diameter can which is set in a 3.0 ft thick reinforced concrete slab. The outer pump case and the associated suction and drain piping set approximately 24 in from the bottom of the outer can after being set into place. After installation, an insulating material (pearlite) was blown into the void between the pump case and the outer can.

A flat steel plate was grouted into the foundation, using epoxy grout, to provide a level surface (0.002 in per ft) for the outer pump case flange to set on. This was performed by using optical alignment equipment as used on large horizontal turbines and compressors.

Originally, the pump was capable of producing a discharge pressure of 2300 psig at normal conditions. However, the downstream system losses were found to be less than expected, so the pumps were destaged by two impellers. The reduction in energy consumption immediately began paying off the cost of destaging. The original equipment manufacturer was consulted as to the best locations to remove the impellers. The pump shaft thrust and the overall rotordynamics were high areas of concern prior to impeller removal. The destaging of the pump improved its overall stability, because the pump was no longer operating back on its curve, but closer to its best efficiency point (BEP).

The motor used to drive the pump is a 1000 hp vertical 4000 volt, two pole, 3600 rpm induction motor with a high thrust load capacity. The coupling, although extensively modified, is a rigid spool design. The coupling hubs were supplied with a slip fit design, though the motor coupling hub was later modified to an interference fit. These pumps were originally commissioned in the summer of 1980. They had been tested at the OEM facility with water at reduced speed. Although initial wear ring rubbing was noted on teardown after the test runs, the pumps were accepted and shipped to the construction site.

INSTALLATION

The design of the installation of any pump system is critical to its longterm operability and reliability. For cold service light hydrocarbon pumps, this is especially critical. The piping system, which consists of suction, discharge, and vent systems, must be designed to minimize pipe strain at the pump flanges and eliminate any possibility of gas pocketing. In the initial design phases, the potential for low temperatures must also be addressed. These low temperatures (−147°F on this system) can be obtained when liquid ethylene under pressure becomes a vapor as it is released to a lower pressure.

The suction tank is a ASTM 516-70 carbon steel, Sharpy tested vessel, which stands 20 ft above grade with an 84 in internal diameter. The height of the tank is 18 ft, 8 in (including the heads), and has a seam-to-seam dimension of 17 ft, 2 in. The liquid ethylene in the tank is at equilibrium and is considered to be a boiling liquid. The orientation and location of the tank was specified to minimize head loss, by lessening the of piping bends. In addition, accessibility for maintenance, particularly head clearance, was considered. The ability to remove a pump can be a challenge if the system is not designed with these elements in mind. The 8.0 in Sharpy tested carbon steel suction piping to these pumps has 0.5 ft of head loss at the design flowrate. At design conditions, the NPSHA is 19.9 ft and the pump NPSHR is 16 ft. All horizontal suction pumping has a slight slope of 1.0 in/10 ft, to prevent pocketing of vapors. Initially, an offset disk block valve was utilized for the main suction block valve at the pump. This proved to be inadequate for isolation purposes, so a second valve (a standard wedge gate valve) was installed with a bleed connection in between, resulting in vastly improved decontamination times.

During periods when both pumps are operating on minimum flow, recycle to the suction tank, a significant amount of heat is put into the liquid returning to the tank. On occasions like this, within one hour the liquid entering the suction of these pumps will begin to boil, and the pumps will vapor lock, resulting in severe damage to both pumps. Operation under these conditions (although rare) has to be carefully controlled. Of course, if it could be justified, a larger exchanger operating off one of the process refrigeration systems could be installed to handle this load.

The vent system (Figure 1) is a critical part of the overall system design. The vent locations are:

- The plan 13 (reverse) seal flush at the seal gland.
- At the seal chamber, below the seal.
- At the discharge of the pump.
- Downstream of the check valve before the discharge block valves.
- Downstream of the discharge block valves.
- 1/2 in stainless steel tubing atmospheric vent, between the pump discharge and the discharge block valves.

All of the vent lines, with the exception of the 1/2 in tubing line, are tied through a manifold to the liquid drain system and to the vapor space of the suction vessel. The tubing is open to atmosphere 20 ft above the pump and used only as a momentary test vent prior to startup.

OPERATION

To prepare a pump for service, the system, usually between the suction and discharge block valves, is purged with nitrogen to

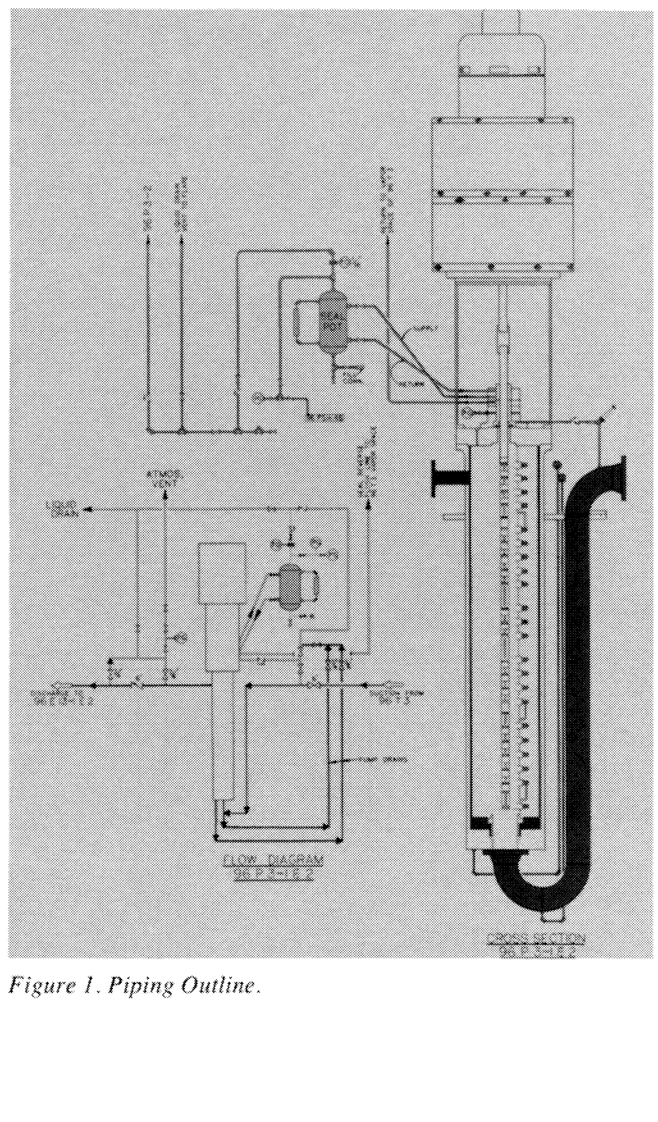

Figure 1. Piping Outline.

atmosphere for a set period of time. Next, the vent valves are lined up to the liquid drain (flare) system, and the pump is flushed with methanol to remove any moisture present in the system. The methanol is then purged from the system with nitrogen through the same line up scheme. Ethylene is then *slowly* introduced to the pump system in order to cool, but, not thermally shock the system or distort any pump parts. After a period of time (approximately 20 minutes), the liquid drain lines are closed, and the valves that line up the vents to the suction vessel are opened. The pump is allowed to set in this mode for a minimum of eight hours. This allows the pump and piping to reach equilibrium with the suction tank and thus, become liquid full. At this point, the pump is ready for startup, and all of the vent valves (except for the seal flush) are closed. The plan 13 flush is at the high point of the pump case and is capable of venting all ethylene vapors generated from insulation losses. Thus, the pump is maintained in a state of readiness. If a startup is to take place, the 1/2 in vent valve is used to check for vapor prior to pump operation.

Guidelines for operation were developed based on previous experiences of other vertical multistage, light hydrocarbon pumps. The sudden depressurization of ethylene from a high pressure will cause thermal shock to the system and potentially a leak. Sudden pressuring of ethylene to a high pressure will generate heat and could cause a rapid and violent decomposition.

When this pump is shut down, every effort is made by the operations personnel to respond immediately to the pump. If the pump is allowed to coast down and the suction valve is not closed prior to its coming to a stop, the pump will rotate in reverse. The stored energy in the liquid from the discharge check valve to the first stage impeller is enough to cause the pump to reverse spin at speeds in excess of 3600 rpm. When this is allowed to occur, the pumped liquid vaporizes and does not supply the heat removal, bushing lubrication, or hydraulic damping that the pump requires. The pump internals are then damaged and when the pump is restarted, the vibration levels are usually excessive and the pump must be removed for a major overhaul. Further design modifications to prevent reverse rotation are being considered; however, many could have a negative impact on the system under certain circumstances. Some possibilities include: a timed motor operated suction valve, an automatic vent on the discharge of the pump, or an antireverse rotation mechanical stop.

When maintenance is required, the pump is isolated, and the ethylene is vaporized and purged from the pump. This is accomplished with warm (ambient) nitrogen and using the liquid drain system.

DRIVER

The driver is a 1000 hp vertical high-thrust 4160 volt induction motor with a 1040 solid shaft design. The radial bearings (upper and lower) and the thrust bearing are in an oil bath, which has cooling coils. A Kingsbury tilt-pad type design is used for the thrust bearing. The radial bearings are a sleeve design with oil grooves cut axially and circumferentially. The upper journal is 8.75 in in diameter, 0.50 in thick and 6.50 in in length (Figures 2 and 3). The lower journal is 4.25 in in diameter, 0.50 in thick and 3.63 in in length (Figure 3). Early on, it was determined that the upper shaft journal would not hold its shape. Each time it was removed, it was found to be 0.003 in to 0.005 in out of round. In addition, it was found that the motor would not run in a stable manner unless a diametrical bearing clearance 0.008 ± 0.00025 in was maintained. If the lower limit was exceeded, the bearing would overheat and the babbitt would melt, which resulted in severe journal damage. If the upper limit was exceeded, the rotor would go into oil whirl. The vibration levels would then continue to rise, as the bearing clearances continued to increase, resulting again in severe damage.

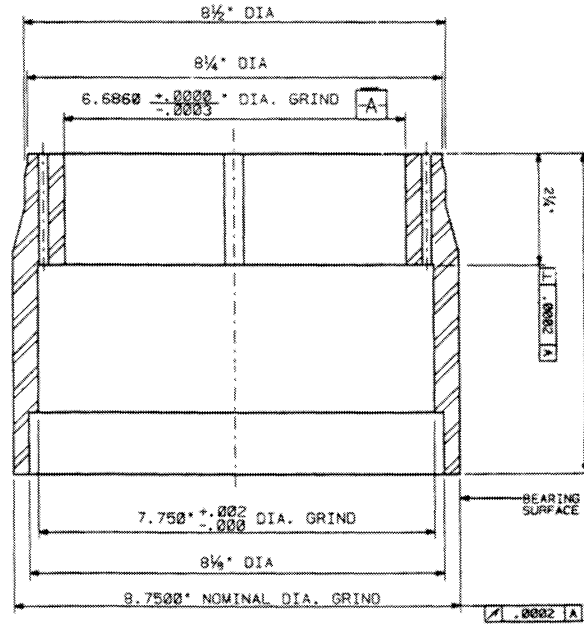

Figure 2. Upper Motor Journal Bearing.

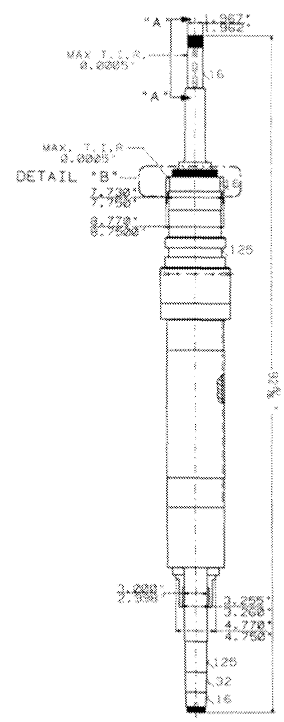

Figure 3. Motor Shaft with Journals.

It was also discovered that the upper section of the shaft was bent after each disassembly. After some research, the mechanism was discovered. The upper part of the motor shaft was stepped down to allow for the oil stand pipe that maintains the oil bath level (Figure 3). When the rotor was raised, using an eye bolt and shackle, then lowered to a wooden cradle, the weight of the armature over stressed the end of the shaft. This resulted in a 0.005 in bend in the shaft. A different lifting procedure is now followed. This involves an additional nylon strap placed around the armature windings and another placed just below the windings. Slowly, the top is lowered while the two other points are lifted, thus eliminating shaft distortion resulting from over stress.

The original oil cooling coils were made of monell tubing. After one and one-half years in service, the coils failed from under deposit corrosion and were replaced, on an emergency basis, with copper refrigeration tubing. This resulted in improved heat transfer and reduced oil and bearing temperatures. Subsequently, a study was made on the compatibility of the copper tubing with the plant cooling tower water. It was found to be compatible, and has proven so.

The motor coupling hub was originally a slip fit with a segmented ring keeper and was key driven. This made achieving and holding the tight alignment tolerances very difficult. One motor shaft was turned down to remove the keyway and to provide a square shoulder for the coupling to stop against. A coupling hub was machined with a 0.0015 in total interference fit to the shaft. The other motor shaft was to be machined in the same manner six months later. However, when the motor hub was removed, a crack was found in the keyway that had propagated about 0.25 in into the shaft and measured approximately 1.0 in long. A new shaft was machined from ASTM 4140, which was prepared to a specification similar to a turbine shaft specification. The coupling end was ground to a 1/2 in per 1.0 ft taper. A 410 stainless steel band was shrunk on the hub to be used as a vibration probe target.

VIBRATION PROBE INSTALLATION AND USE

To the authors' knowledge, no one at the time (1980) had truly instrumented a vertical motor/pump with a permanent vibration and thrust monitoring system. Engineers at two other Gulf Coast plants were experimenting with accelerometers, but did not have enough operational data to give a precise direction for vibration monitoring.

Two proximity probes were mounted at the top of the motor referencing the motor shaft position axially. Additionally, two radial probes were mounted on the outboard (top) of the motor located just above the shaft bearing journal. Two radial probes were mounted below the pump coupling hub on the seal gland, reading the vibration of the pump shaft.

Initially, a tri-accelerometer was mounted to the bottom of the motor. The vibration detected by the accelerometer did not give a true representation of shaft vibration in a manner that could be used for problem identification. It was later replaced with proximity probes. To gain further knowledge and understanding of the pump, another set of radial probes were installed and referenced the motor shaft at its coupling hub. These probes proved to be extremely valuable. It was then discovered that the two probes mounted on the pump seal gland were often at or near a pump shaft vibration node. The only time the pump shaft vibration levels were accurate was when the upper pump bushing clearance was excessive and the pump was rubbing internally, or from misalignment of pump and driver. Otherwise, the vibration readings taken at the top of the seal gland were misleading. Because the motor is rigidly coupled to the pump, the new vibration probe mounting gives a true vibration reading of the rotor system. If this had been a flexible type coupling design, then the vibration readings would not have been a true indicator of the pump shaft vibration.

The motor is soloed (run uncoupled) after any maintenance work to the motor or pump that requires uncoupling. The vibration levels are monitored and compared to other solo information previously taken on the same motor. Normal levels are about 1.0 mill peak to peak (pk-pk). The key to the solo is the early detection of bearing problems in the motor and particularly oil whirl.

After the pump has been coupled to the motor and the system has been readied for operation, the pump is started on minimum flow recirculation to prove out the work that has been performed. At this time, the mechanical maintenance technicians are graded on their work. Usually, the technicians are at the vibration monitoring panel watching the startup. An amplitude of 3.0 mills pk-pk is the best that has been achieved on a coupled system. The technicians know they have done an excellent job if the vibration levels are below 5.0 mills pk-pk. Since they have been working on one of only two product pumps around the clock, they know how critical these pumps are, and the self satisfaction of doing an excellent job is readily apparent.

After maintenance is performed, the pump startup is monitored using an oscilloscope, spectrum analyzers, and a digital vector filter. The expected vibration level is 5.0 to 7.0 mills pk-pk at the motor shaft coupling hub. With the old coupling design, a vibration level of around 10 mills pk-pk was considered very good.

It has been demonstrated that the pump will run indefinitely at a level of 12 mills pk- pk. Above 12 mills pk-pk, the pump is scheduled for overhaul in the near future. At 18 mills pk-pk, the pump must be shut down and repaired prior to major pump and motor damage.

While the pump is on minimum flow recycle, the vibration levels are typically 1 to 2 mills higher than when the pump is fully loaded. This can be attributed to the flow instability of this type of pump, when it is operated back on its curve. Originally, the minimum flow was sized for 280 gallons per minute (38 percent BEP). This recycle rate was found to be inadequate for long term runs on recycle in excess of *one* hour, and has been increased to 400 gallons per minute (55 percent BEP).

The alarm level on the proximity probes is set at 12 mills, and the danger is set at 18 mills. The radial vibration alarms are audible only and are not interlocked to a shutdown. A low level in the suction tank and the axial position of the motor rotor (thrust), are the only two system monitors that are interlocked to shut down these pumps. When the pump is shut down, either manually or by an interlock such as the suction tank low level, two small bayonet type electrical heaters are automatically turned on. This is to prevent an ice ball from forming around the top side of the seal gland, onto the pump shaft and coupling. The two heaters are mounted in a two-piece aluminum clamp that is installed prior to the vibration probes. Once the vibration probe brackets are set and the vibration probes gaped and connected to the oscillator demodulator, access to the coupling area is restricted.

PUMP PARTS DESIGN

The design of this pump is a good design, and probably the best choice that could have been made for the ethylene service in which it operates. However, one major problem with metallurgy was encountered and a few lesser problems which affected the long term reliability and maintainability of the pump. Initially, the pump was provided employing the extensive use of 18-8 stainless steel clearance parts throughout the pump. As a result, both pumps seized on initial startup when rubbing contact was made between these parts. Also, as of the late 1970s, the manufacturer of this pump was not performing a dynamic assembly balance on this style of pump, because the design did not lend itself to dynamic assembly balance. Only the impellers were balanced, on an individual basis, before assembly. Finally, the shaft, which was made of monel K-500, did not always remain straight throughout a run cycle.

During the initial startup of these pumps on ethylene process, serious mechanical problems were encountered. As each pump was started, it very quickly ran into problems unrelated to the design, and had to be shut down within seconds. During coastdown, each pump seized to the point that it could not be turned, even with the assistance of an eight ft bar. At teardown, the four interstage bushings, throat bushing, and most of the impeller eye and hub wear rings had been severely rubbed, as were the case wear rings.

The throat bushing and center case bushing had Ni-Resist II metallurgy, while their mating sleeves were flash chrome over 18-8 stainless steel. They both had design clearances of 0.008 in/0.010 in. The remaining three interstage bushings were made of bearing grade carbon with mating sleeves of flash chrome over 18-8 SS, and running clearances of 0.016 in/0.018 in. The clearance diameter for all five bushings is 3½ in. Each impeller case ring was made of 18-8 SS with flash chrome applied to the bore. Each of the impeller eye and hub rings was of 18-8 SS metallurgy which had an "electrolyzed" hardening process applied to the OD. The impeller eye rings, with a clearance diameter of 6⅛ in, have a design clearance of 0.014 in/0.016 in. The impeller hub rings, with a clearance diameter of 4⅝ in, were designed to run with a 0.012 in/0.014 in clearance.

As time was important, the startup team agreed to the use of SAE 660 leaded bronze (ASTM B584 composition UNS No. C93200) as the material to mate against the 18-8 SS parts, in spite of concerns that copper acetylides can be formed in ethylene streams if acetylene is present in the stream. Most engineers feel that the chance of this phenomenon occurring in a nearly pure ethylene process in the absence of corrosive media is remote. Positive plant experience with the bronze material, along with experience canvassed from other petrochemical plants, led to the decision to forge ahead. Bronze tends to behave like an abradable material in applications like this and has a high thermal conductivity. Thus, it can absorb the momentary impacts that take place during startup, shutdown and occasional upsets. During the repair of the initial startup failures, the impeller wear rings were replaced with the bronze material while the throat bushing was bushed with bronze. Other materials remained unchanged. At the next repair, the other four interstage bushings were also changed to bronze [1].

Over the next seven years, the material combinations remained the same, although under high stress conditions, the bronze was observed to wear rapidly to the point that the rotor would exhibit an ever increasing vibration amplitude. In an attempt to extend the life cycle of the wear parts, the rotating parts were changed to Nitronic 50 and the stationary parts to Nitronic 60, both 300 series stainless steel products, specially formulated for low galling tendencies. This combination had been used successfully in other applications at the plant on some horizontal multistage pumps, also in cold, light hydrocarbon service. Initial results appeared positive, however, in mid 1990 the pump, during a motor bearing failure, seized twice, causing approximately $20,000 worth of shop repairs in each case. The original bronze alloy was used once again, this time only in the interstage bushings and the first stage impeller wear rings. It appears that when the motor bearing clearance becomes excessive or during process upsets, starts and stops, the rotor system seems to pivot in a way that imparts a heavy load on the lower wear parts of the pump. This was evidenced on many failures, as the lower bushing and the first few impeller wear rings would be severely worn and distorted, and the damage would become progressively less as one moved toward the upper part of the pump. As a result of these experiences, the authors feel that any future material changes to these parts will probably involve a nonmetallic material for the stationary impeller wear rings and the interstage bushings. Also, a modification to the pump case may be required to provide a higher volume of liquid to cool and lubricate the lower interstage bushing.

From the early stages of the olefin unit project, most of those involved were in favor of having a dynamic balance performed on this 20 stage rotor. However, such a balance could not be performed because the balance stand rollers could not be placed properly at the lower end of the shaft. In order to balance the rotor with the existing design, one would have to assemble the rotor once for balance, without the stationary rings in place, then disassemble and restack the rotor with all parts in place. This would be time consuming. Moreover, the risk of upsetting the balance would be real due to the amount of heat required to disassemble the rotor, which often results in impeller bore distortion. In order for an assembly balance to take place without an ensuing disassembly, the four interstage bushings would have to be redesigned, so as not to interfere with the balance operation and to allow placement in a balance stand.

The interstage bushings and most of the impeller hub case rings were converted to a split design (Figure 4). The impeller hub case rings were converted in order to aid the balance operation and to eliminate the possibility of distortion of both hub and eye stationary rings during disassembly. Plant engineers worked with the pump manufacturer to design split bushings for a pump which was not originally designed with this in mind. Split bushings must be designed in a way that ensures the original integrity of the solid bushing and does not allow fasteners and alignment pins to loosen and fall into the pump during operation. The key to successful manufacture of split rings and bushings is to prepare the mating surfaces initially, with alignment pins installed, then clamp the halves together. The part should then be match marked and finished as a whole part. A valuable secondary benefit of this design is that it allows accurate runout checks on the assembled rotor. However, in order to obtain good runout data, the interstage sleeves where the rollers are to be placed must have concentricity and runout limited to 0.0005 in.

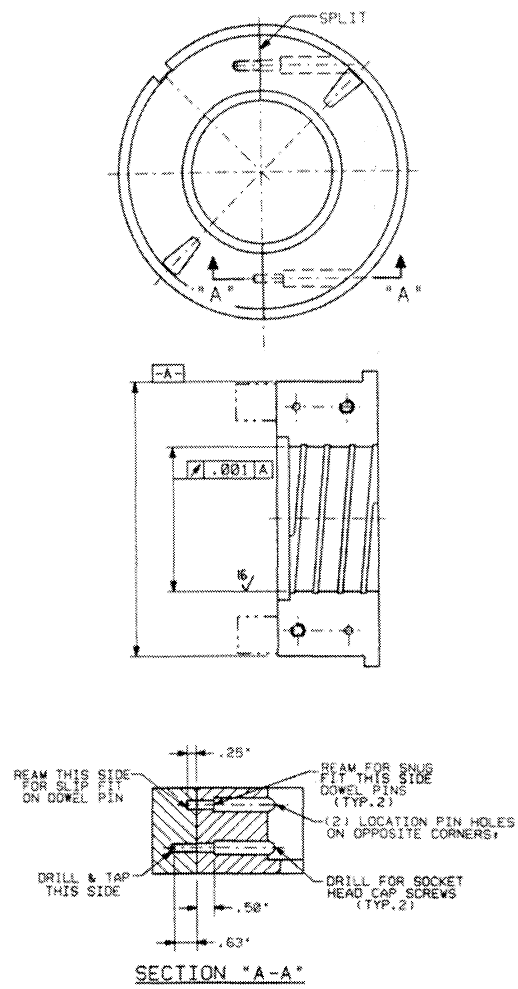

Figure 4. Split Bushing.

The balance sequence for the rotor starts with an individual balance of all impellers to the, API 610 7th edition, 4W/N tolerance. In the balance stand (Figure 5), the rotor is set with rollers placed at the second interstage sleeve and at throat bushing sleeve. The impeller eye case rings and the two bypass cylinders are held away from any moving parts with tape supported from a piece of pipe. The balance operation then proceeds working from the middle out until the 4W/N tolerance is met. On this 500 lb rotor, this translates to 7.9 gram-in per plane.

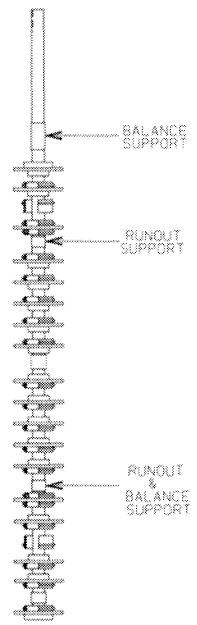

Figure 5. Pump Rotor.

During the first few years of operation, the shaft displayed a nagging tendency to distort while in operation and during maintenance. Impeller and sleeve removal was also very time consuming and often damaging to several of the parts due to the heat and force required during this operation.

The first attempt to solve these problems involved the use of 17-4 PH condition H- 1075 precipitation-hardening stainless steel as the shaft material. This material had improved tensile strength and a much higher yield strength than the cold drawn, age hardened monel K-500. Also, in an attempt to lessen the damage to the shaft and fitted parts during disassembly, the impeller and sleeve fit surfaces had a 0.012 in/0.015 in finished thickness coating of flame sprayed chrome oxide applied. During manufacture, the shaft had to be straightened four times, but the finished product, complete with coated surfaces, maintained a TIRR of 0.002 in throughout its length.

The shaft proved to be a success, requiring much less effort at disassembly and resulting in 60 percent fewer impellers requiring bore reclamation. The shaft retained its straightness, which reduced repair time and cost. The engineering team felt that this was a major contributor to the improved vibration levels that the motor/pump system was demonstrated.

Finally, based on recommendations of metallurgists, the shaft material was changed once more to 15-5 PH condition H-1100 material to improve the machining process. This resulted in less distortion and fewer straightening cycles. The change to the H-1100 heat was intended to improve the shafts low temperature impact values. However, it also resulted in a sacrifice of tensile properties, but has continued to prove superior to the original material [2].

MECHANICAL SEAL

The original design specifications for this pump requested the use of a double mechanical seal. The seal was a standard pusher type seal with an unbalanced inner seal and a balanced outer seal. The seal system (Figure 6) employed the use of a reservoir which was outfitted with an elastomeric diaphragm and a 50 lb plate which was placed on top of the diaphragm. The lower side of the reservoir beneath the diaphragm and the volume between the two seals was inventoried with ATF fluid to a predetermined level. The top side of the diaphragm was referenced to the process side of the seal and had a small amount of methanol placed on top of the diaphragm to act as a lubricant and to provide a buffer between the process gas and the diaphragm. As a result, the inner seal carried a steady state load of 50 lb, while the outer seal carried the load of the process pressure plus the 50 lb weight, approximately 400 lb. A process recirculation line was provided at a point just above the top guide bushing (throat bushing) and directed back to the suction of the pump. Its purpose was to provide a continuous flow across the bushing and to maintain a stuffing box pressure approximately 75 lb above suction pressure.

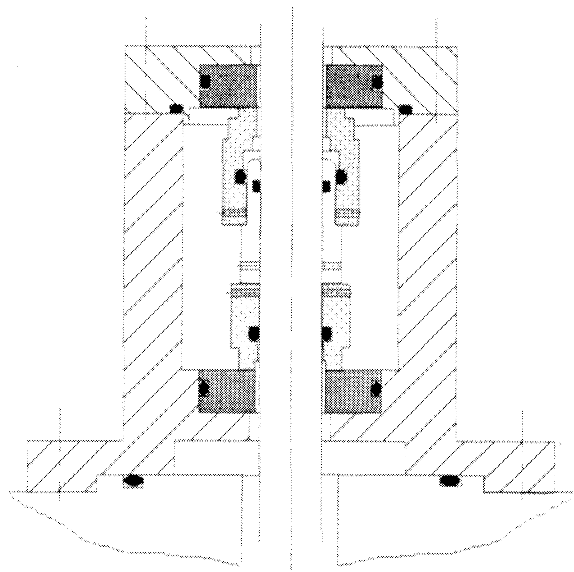

Figure 6. Double Seal Arrangement.

The pump manufacturer claimed that this design had been used in many cryogenic applications with enormous success. However, in this case, the process temperature is a "mild" –37°F with the pump located outdoors on the hot Texas Gulf Coast. As a result, the outer (load seal) produced so much heat that the seal was failing on a monthly basis. The ATF was replaced with an ethylene glycol and water solution which resulted in a minor improvement. A makeshift cooler and circulating pump had to be added to alleviate the problem. Also, during preparation for startup, the inner seal mating ring was often dislodged by the sudden introduction of nitrogen and methanol. Attempts to pin the ring and modify the "air free" procedure were made with little success. It was obvious that this particular design had been misapplied. In addition, this setup, as designed, created other problems as follows:

- Bleed off of top section of pump prior to startups was a manual operation, which was wasteful and time consuming.

• The seal design could be easily inverted, thus creating a potentially disastrous situation.

• The diaphragm would fail periodically,showing the same symptoms as a seal failure; the pot had to be inspected at each seal failure.

• The recirculation line was introducing warm gas to the pump suction, which is a detriment to a light hydrocarbon pump riding on a narrow NPSH.

The plant personnel surmised, based on past experience, that a tandem seal system would be far superior in this case and also could be designed in a way that would address the other problems (Figure 7). The tandem seal design, also a pusher type, required a new seal chamber design with a larger bore and a standard seal pot. The new chamber was designed to be lighter (to facilitate an improved alignment technique) and to provide the seal with a hospitable environment in which to operate. With these changes, seal orientation would no longer be a problem and the presence of a weighted elastomeric diaphragm would no longer be required. Additionally, a plan 13 flush arrangement was incorporated, which was routed back to the vapor space of the suction vessel, thus providing adequate cooling to the seal. Also, the pump would remain liquid full, via gravity flow, even when on standby. Care was taken not to pocket this line in order to encourage a continuous cooling flow during standby. The recirculation line is no longer used, and the pump remains ready to start at any time without bleeding off valuable product from the inner pump case.

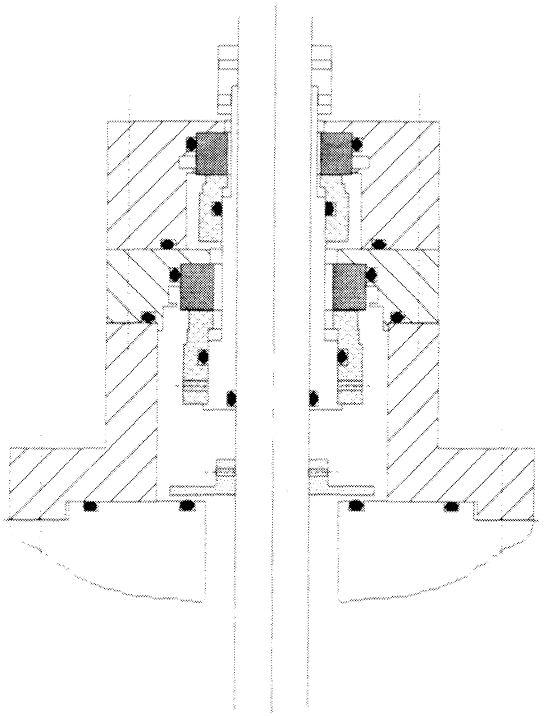

Figure 7. Tandem Seal Arrangement.

In addition to solving the problems directly related to the seal system, a way to shorten the length of time to decontaminate the pump for seal changes was pursued. Normally, a unit processor would have to depressurize, then blow warm nitrogen through the pump for hours before a vapor analyzer would show the system to be safe. After a seal change, the processor would have to "air free" and dehydrate the system using methanol, then blow the methanol out with nitrogen before ethylene process could once again be introduced to the pump. A back sealing device was designed ("static seal"), so that during seal changes, the pump would only need to be depressurized, and the remainder of the procedure would no longer be required. The design involves a small collar which is fastened to the pump shaft just above the upper cover. When the shaft is disconnected from the coupling, the collar sets on an O-ring installed in the upper cover. The collar is set in place against the O-ring, when the rotor is located approximately 0.015 in off bottom. This setup has proven very successful, safe, and is a tremendous time saver.

The changes in the seal system have provided the plant with a system which features simplicity and reliability. Life cycles for the new seal system have been in excess of four years.

COUPLING DESIGN

The coupling for this pump is a rigid type hub and spool coupling with a lift nut for setting the pump rotor position in the pump (Figure 8). This type of coupling has been a standard for applications such as ours for many years and has a good record throughout industry. However, the authors recognized after several long arduous alignment sessions, that perhaps the coupling itself could be the cause of most of their alignment woes. After all, they have gone to great lengths to ensure the straightness of the pump and motor shafts. What else could make the rotor system appear to be bent or bowed sometimes, but not other times? Great care was taken to carefully machine the coupling during repairs and during occasional manufacture without much improvement in performance.

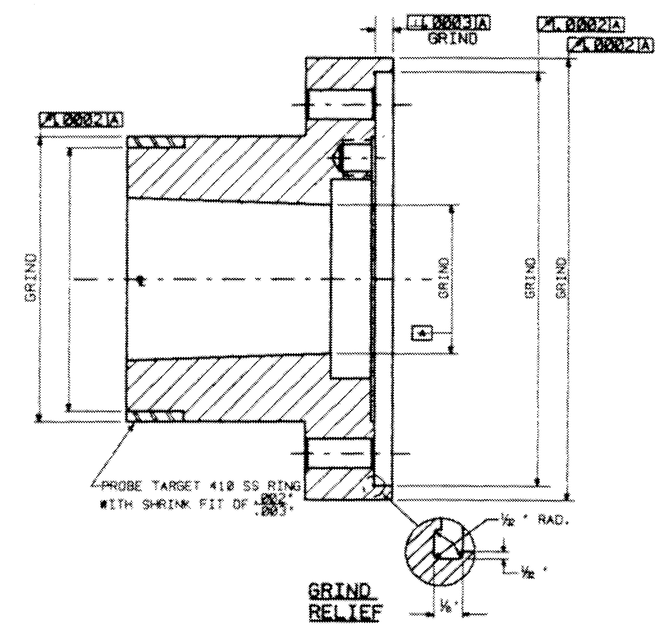

Figure 8. Coupling Assembly on Mandrels.

An analysis of the coupling design was made resulting in the realization that the perpendicularity, concentricity, and parallelism of the mating surfaces should be of utmost concern. A quick look at the effects of runout on the six mating faces on this coupling is helpful. The six mating faces on the coupling were each being manufactured within 0.001 in/0.002 in TIRF. Assuming a rigid system, for simplicity, one could expect a 0.174 in offset at the end of the 12 ft pump shaft. In reality, this is reduced significantly by rotating the parts with relation to one another until a "sweet spot" is found. However, this made coupling assembly lengthy and often the precise results desired were not obtained. This led to the realization that either another type of coupling or an improved version of the existing coupling should be used.

Other coupling styles were considered but, in the end, the plant elected to stay with the existing style with some improvements. The problems which needed to be addressed were;

Rust of the 4140 HT components. While the 4140 HT in use did not rust heavily, it still created installation problems and made dismantling a sometimes damaging operation. Clean up of the fits often resulted in out of tolerance fits (probably more than the engineers realized at times).

Radial fits. The motor shaft coupling hub was designed as a 0.0005 in/0.0010 in slip fit. While this is convenient, it added slop to the coupling assembly that encumbered alignment. Other fits needed to be tightened.

Bolt size/arrangement. Torquing of the 3/4 in coupling bolts was nearly impossible in the confined space which the technicians had to work. Also the four bolt design used reduced the possibilities for finding the "sweet spot" during coupling installation.

Radial and axial runouts. As mentioned above, the symmetry of the assembled coupling is critical, and the design would have to address this.

Addressing the rust problem is not the simple problem that it first appears to be. Strength, stability, and resistance to galling also have to be considered in the material selection. For this case, 4140 HT would continue to be used for the motor and pump hubs, while the spool material was changed to 17-4 PH. The pump hub is required to slip on the pump shaft with minimal clearance. Thus, a material which will not gall on the 15-5 PH shaft was needed. The lift nut, while still using 4140 material, is coated with 88-12 tungsten carbide-cobalt on the mating faces and on the OD, which registers on the pump hub and the spool. The coating provides a surface which is rust free and very hard. The hard surface is desirable in that it can be ground to very exacting tolerances and resists wear (from handling). Furthermore, it provides a non galling surface for the 4140 pump hub and the 17-4 PH spool, which it mates with, as well as the 15-5 PH threads on the pump shaft.The radial fits for the four major parts of the coupling are made as tight as practicable. As mentioned earlier, the motor hub was resized having a shrink fit to the motor shaft of 0.001 in per inch of shaft diameter. Thus, any slop in this position was eliminated. The pump hub must be able to slide on the pump shaft but, have as little slop as can be managed. A compromise is in order here; therefore, a maximum diametrical clearance of 0.0005 in is maintained in this position. The three 7¼ in registers on the flanged parts of the coupling also require the same compromise. However, a wider clearance has to be used. This is due to the difficulties encountered at assembly, which the short registers and larger diameters create. A diametrical clearance of 0.001 in/0.002 in is used in this position and creates minimal runout problems at assembly due to the lack of a key at these fits. If necessary, the pieces can be centered at assembly.

To solve the bolting problem would require bolts of smaller diameter and more of them at the two locations. The four ¾ in bolts require a torque in the 211 ft-lb range to obtain a clamp load equivalent to eight ½ in bolts of the same material at 100 ft-lb. The resultant clamp load of 95,200 lb results in the ability of the coupling to carry a torque of 4,808 ft-lb based on friction. This leaves ample safety factor for the normal load of 1,458 ft-lb and the maximum torque of 2,917 ft-lb. The design loading is based on friction because of a "loose" fit in the mating members. If bolt loading were used the torque capability of the coupling would be 9020 ft-lb. This has proven to be the proper solution to the torquing problem and also allows for more latitude when fine tuning the installation, i.e., the "sweet spot."

The method used to determine the torque capability is as follows [3]:

Solve for clamping force provided by the bolts.

$$C = .8 \times S_y \times A_t \qquad (1)$$

Solve for frictional torque capacity.

$$T = C \times u \times R_f \qquad (2)$$

Where the friction radius is;

$$R_f = \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right) \qquad (3)$$

Check for bolt shear loading, assumes equal loading (body bound design).

$$T = S_s\left(\frac{\pi \times d^2}{4}\right)\left(\frac{D_{bc}}{2}\right) \times n \qquad (4)$$

The manufacture of the coupling components is a straight forward process but, requires attention to detail. Partial drawings shown in Figures 8 and 9 show limits of perpendicularity of 0.0003 in, parallelism of 0.0002 in and concentricity of 0.0002 in. Note the

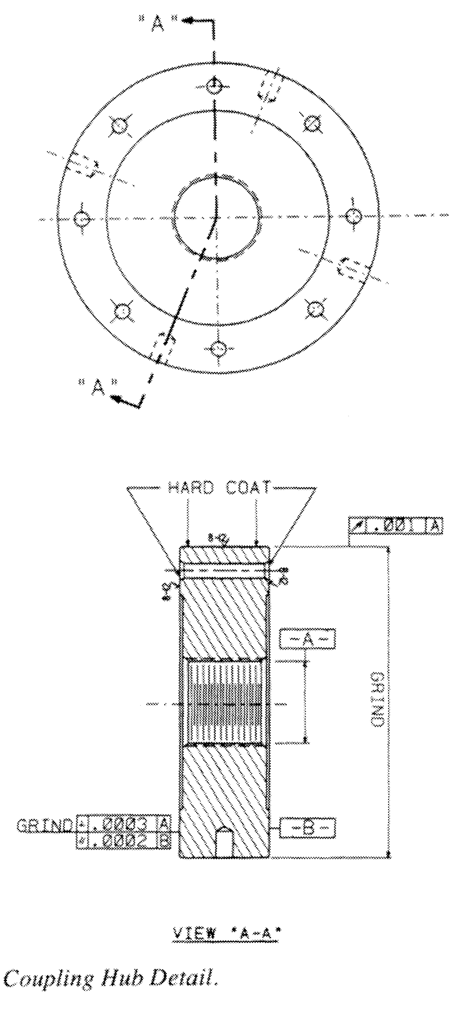

Figure 9. Coupling Hub Detail.

use of grind reliefs and chamfers in the design, which aid in manufacture of the coupling to these tolerances. The bolting design does not include a body bound assembly. However, it does result in a close tolerance fit, which does not create unacceptable balance drift during subsequent assembly operations. After manufacture, the coupling is assembled and checked for runouts. Corrections are made as required at this time.

Assembly of the coupling at the shop requires attention to detail, and for this, a detailed procedure was developed to follow, of which some of its points will be discussed here. First, some pitfalls to avoid:

- Keep the surfaces immaculate.
- Watch for and avoid surface scratches, digs, etc.
- Avoid excessive and uneven heat application.
- Remember to install the half keys for balance purposes during assembly.
- Bolts should not bind.
- Lift nut should not effect runout in any orientation.
- Assure that bolt and nut weights are matched within 1/10 gram, not matched to the coupling.
- Do not grind off old match marks; stamp an "X" over them.

Assembly of the coupling for check out and balance requires the use of a set of precision mandrels. The mandrels that are used have been coated with chrome oxide in the fit areas to ease the removal of the hubs and to reduce the chance of scuffing or galling. The mandrels for both pump and motor hubs allow for a 0.0005 in to 0.0020 in shrink fit. A detailed check of each phase of the assembly is made using a diagram and check list. In all, four different diagrams are used and seven sets of data are taken during the check out process. The fourth diagram (Figure 8) is used four times while

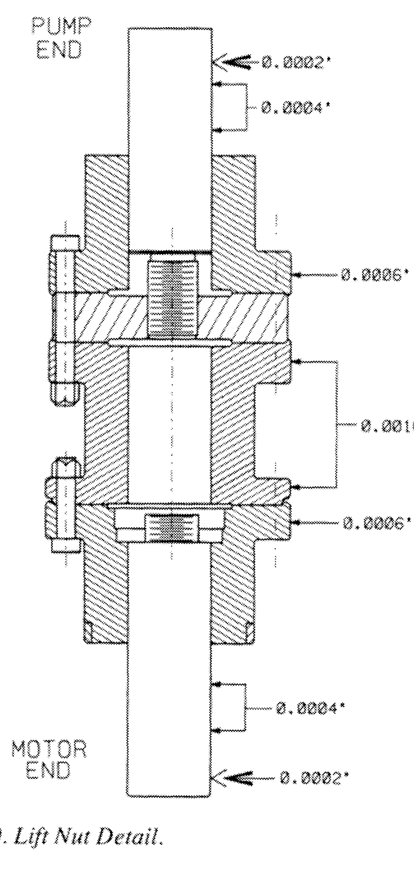

Figure 10. Lift Nut Detail.

taking runout readings with the lift nut set at 90 degree intervals through one turn. These extra steps virtually assure a successful field installation. The maximum runouts shown must be met or rework will be necessary.

The balance operation begins before the assembly steps with an individual balance of the four major components. They are all single plane balanced in accordance with ISO G-2.5. This operation is normally performed using a magnetic mounting plate in lieu of Mandrels, which impart inaccuracies to the balance operation. After assembly of the coupling on the two mandrels, the entire assembly is moved to a balance stand where a two plane balance is performed also to ISO G-2.5. The lift nut must not have its balance altered during the assembly balance, as it is required to be free of match marks, allowing location at any of eight 45 degree locations. Therefore, while the lift nut is present in the assembly balance, no weight correction can be made on it. After the assembly balance, match marks are placed in such a way as to orient the two hubs and the spool identically during future assembly.

ALIGNMENT

Alignment of a critical, vertical process pump starts at the foundation. The mounting plate, when grouted into place should be level within 0.002 in/ft. The mounting flange on the "can" must provide a surface for the pump case to sit on, which will maintain the level provided by the foundation (Figure 11). This requires precision machine work at the factory as well as conscientious care of these mating surfaces in the field by the users technicians.

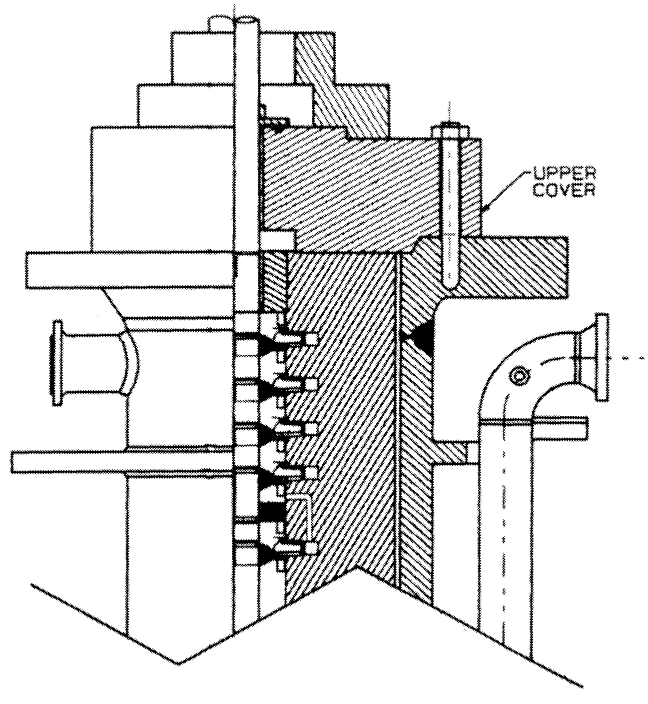

Figure 11. Pump Mounting Cross Section.

Mounting of the upper cover to the pump is a very critical step in the assembly of the pump, which can have a dramatic effect on the alignment of the pump. With the pump case having been properly line bored at the factory, the fully dressed rotor will, ideally, run on the center line of the case. However, the upper cover can change the reference to this center line if it is not a precision component, or if it is not properly cared for. Because the cover must be installed on the pump case after rotor installation and before installation in the field, the inner bore and upper face on the cover become the primary reference points for all alignment operations. This means that the cover cannot be field aligned, but must register perfectly to the pump case as installed. A natural reaction to this realization is to expect a nearly perfect relationship between the cover and the pump. In reality, economics and the capabilities of a precision machine shop will determine what tolerances are acceptable. To determine the status of this relationship, the cover was mounted on the pump case with the rotor removed, in order to access points of reference in the case bore. At a precision facility, the axial and radial runouts of the upper cover, pump case register and cover/case assembly on all three pumps were checked. Radial runouts were as high as 0.008 in TIRR and axial runout as high as 0.002 in TIRF. Improvement was indeed needed and possible, the goal being to obtain a proper fit up of these two heavy and bulky parts as near perfect as practicable. An allowable diametrical clearance of 0.001 in/0.002 in at the register fit, between the cover and the pump, was settled on and resulted in successful installation. These clearance and machining tolerances on the cover result in an actual radial runout of 0.0015 in/0.0025 in on the cover bore with relation to the pump bore. More critical is the parallelism of the two mating surfaces or faces of the upper cover. A precision grinding facility was used to obtain parallelism between the two mating surfaces within 0.0005 in, resulting in a finished relationship to the pump bore within 0.001 in. These tolerances eliminate the need to resort to "shimming" or other questionable techniques to obtain acceptable alignment.

With the upper cover mounted to the pump, the motor mounting bracket is installed and secured to the pump assembly. The motor is then installed with utmost attention to cleanliness at the fit surfaces. At this time, motor to pump alignment is performed by mounting a dial indicator to the motor coupling hub and indicating off the upper cover bore (0.001 in TIRR) and the upper cover face (0.001 in TIRF) on a 4.0 in radius (Figure 12). The preceding step was made possible by two modifications, which allow the installation of the seal chamber to take place after installation of the motor. First, the seal chamber was made much lighter by developing a more compact design, thus making installation possible without the assistance of an overhead lifting device. Also, in order

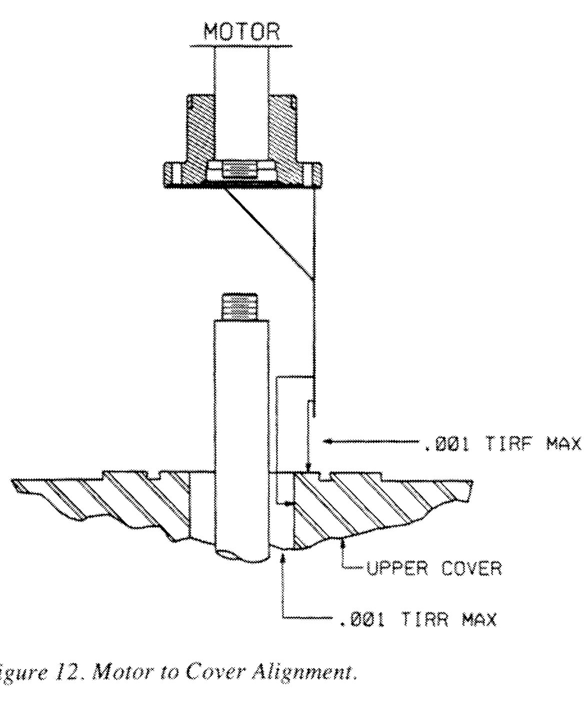

Figure 12. Motor to Cover Alignment.

to fit between the two shaft ends, the pump shafts had to be shortened by 3¼ in.

The static seal and seal chamber are installed taking care to align the seal chamber to the pump assembly. This extra step is taken in order to allow future alignment checks using the seal chamber as a reference, thus bypassing removal of the lighter but still heavy seal chamber. Also, because of the presence of the static seal discussed earlier the pump needs only to be depressurized, not decontaminated if the chamber is left in place.

The coupling is now installed using a very meticulous procedure which was developed at the plant. The authors' experience has been that the pump manufacturers stress good alignment, but do little to assist the user in developing good procedures which are applicable to the plant environment. Also, factory alignment for test stand purposes is accomplished using equipment often not available in the field and usually performed by "old timers" who carry the procedures in their heads. The rigid coupling design used on this pump has eight mating surfaces that must be in near perfect condition in order to obtain an acceptable long lasting alignment. The rigid coupling, in effect, makes the pump and motor rotors behave as one rotor. Therefore, any distortions introduced at the coupling will essentially kink the rotor system. All parts must be very clean, protected and have passed a detailed visual inspection. If in doubt, a detailed mechanical inspection is made. Installation begins with the motor hub being fit to the motor shaft with a shrink fit. A slip fit was used by the manufacturer but, good alignment often was difficult to obtain and shaft fretting was a problem. The motor hub is normally installed at a repair facility, usually after an overhaul, where runouts and fits can be checked and corrected as needed. After any installation, field or shop executed, hub runouts are checked (Figure 13). The motor in this system, having sleeve radial bearings and a tilting pad thrust bearing, is susceptible to "shifting" during rotation of the shaft. To determine if this is occurring, it is useful to have indicators both on the shaft and the hub during the runout check. Next, the pump hub and lift nut are positioned on the pump shaft. The coupling spool is then installed taking care to align the balance match marks. After evenly tightening the bolts to the prescribed torque values, another set of runout readings is taken (Figure 14). The total rotor float is checked at this time, and the "lift" is set within tolerance. Finally, the pump hub is connected to the spool while aligning match marks between hub and spool. Another set of runout readings is taken on the pump hub and shaft (Figure 15). Assuming that the previous alignment and coupling checks are accurate, out of tolerance runout readings at this time usually signify a bent shaft. If acceptable readings cannot be obtained, the problem is sought and found. Shimming or selective torquing of the coupling is not permitted. A final set of readings are taken and must be within the maximum values shown.

A.	Motor Coupling Hub Flange	0.0010" TIRR
B.	Top Spool Flange	0.0010" TIRR
C.	Bottom Spool Flange	0.0015" TIRR
D.	Pump coupling Hub Flange	0.0015" TIRR
E.	Probe Target	0.0020" TIRR
F.	Pump Shaft	0.0020" TIRR

CONCLUSION

The authors of this paper have conveyed experiences, on one pump application, that were developed over 11 years of operation. From the beginning, this pump application was recognized to be critical and mechanically challenging. Their belief is that some of the problems could have been avoided up front if better communication within their company and with outside contacts had been

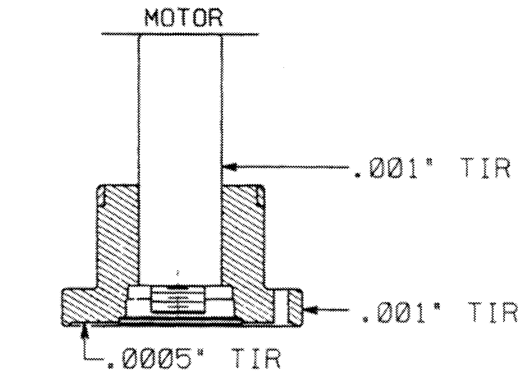

Figure 13. Motor Hub Runout Checks.

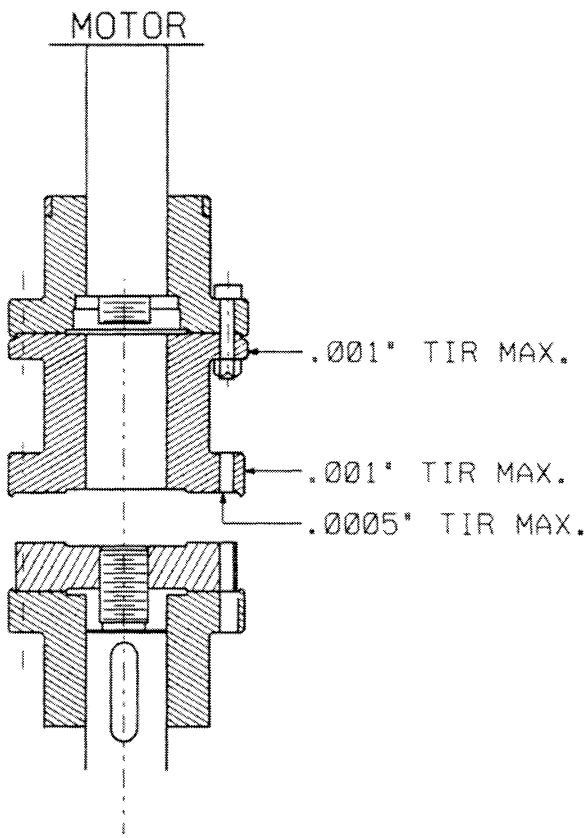

Figure 14. Spool Runout Checks.

established during the design phase. The experiences that were discussed led them to believe, firmly, in up front involvement of plant equipment engineers in the design process.

NOMENCLATURE

A_t Stress Area (in^2).
C Clamping Force (lbs).
d Nominal Bolt Diameter (in)
D_{bc} Diameter of Bolt Circle (in)
n Number of Bolts
R_f Friction Radius (in)
R_i Inside Radius of Mating Surface (in)

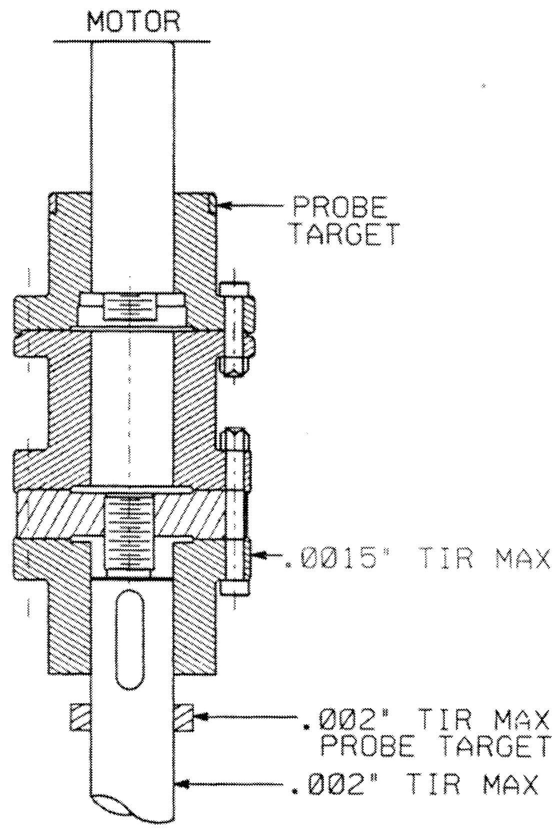

Figure 15. Pump Hub, Target, and Shaft Runout Checks.

R_o Outside Radius of Mating Surface (in)
S_s Allowable Shear Stress (PSI)
S_y Fastener Yield Strength (PSI)
T Torque Capacity (in-lbs)
μ Coefficient of Friction

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