Proceedings of the

15th TURBOMACHINERY SYMPOSIUM

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PROCEEDINGS OF THE FIFTEENTH TURBOMACHINERY SYMPOSIUM

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Texas A&M University

Dr. Dara W. Childs, P.E., Director

Dr. Jean C. Bailey, Editor

with assistance from

The Advisory Committee

and

The Staff of the Turbomachinery Laboratory

November 1986

PREFACE

These *Proceedings* contain papers from the lectures, tutorials, and a special paper for the Fifteenth Turbomachinery Symposium, held in Corpus Christi, Texas, 11-13 November 1986. The Symposium is sponsored by the Turbomachinery Laboratory of the Department of Mechanical Engineering, Texas A&M University.

The Turbomachinery Symposia were established as a forum for manufacturers and users of industrial turbomachinery. Because of many overlapping areas in interest, the Symposia are directed primarily to commercial users with the utility and petrochemical industries.

The Advisory Committee for the Fifteenth Symposium and past symposia have had a continuing influence on the content and direction of the symposia. The committee is composed of recognized leaders in the commercial turbomachinery field from manufacturers, users and universities. Based on their experience and knowledge of the field, papers are solicited and selected to address contemporary problems of interest. Their continued assistance is wholeheartedly appreciated.

Essential elements of the symposia which are not covered by this proceedings include two, one-day short courses which precede the symposium, six discussion groups, a panel session on "Origin and Applicability of API Standards," and a product exhibit show. The short courses are: 1) Fundamentals of Turbomachinery Performance, and 2) Turbomachinery Basics.

The discussion groups are led by engineers with a great deal of experience in the subject areas, who facilitate discussion from the floor. Attendees *actively* participate in the discussion groups, and many use the discussion groups to get sound advice for their peers on problems of immediate importance. The discussion groups facilitate a transfer of information across industry boundaries.

The product exhibit show has over ninety representatives and features new products, accessories and analysis tools. This aspect of the symposium has continued to improve over the past several years in the quality and range of of products exhibited.

Again, the vigorous support of the Advisory Committee is very much appreciated. My very considerable thanks are also extended to authors,, turorial leaders, panel members and discussion leaders. Finally, the efforts of the Turbomachinery Laboratory staff in seeing through the detailed execution of the Symposium are greatly appreciated, with particular thanks extended to Mignonne Jakus and Dorthi Bearden. With regard to this proceedings, a special thank you is extended to Dr. Jean Bailey for her work in editing and preparation and to David Hassinger for his engineering assistance.

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LECTURES

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TESTING AND STARTUP EXPERIENCE WITH ROTATING EQUIPMENT AT NATION'S FIRST COMMERCIAL SCALE COAL GASIFICATION PLANT

by

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Mr. Mohan spent the first four years of his engineering career at Dresser Clark in Olean, New York. He was involved in rotordynamics work and did extensive analysis and testing of bearing and seal designs for high pressure barrel compressors. For the next six years, he was with Exxon Chemical Company and provided consulting assistance on troubleshooting, equipment upgrade studies and new equipment design audits. He was involved in the startup of Exxon's Baytown Olefins Plant and has done extensive work in online/computerized monitoring of critical machinery.

Mr. Mohan received his B.S. degree in Mechanical Engineering from I.I.T. Madras, India, and an M.S. degree in Mechanical Engineering from the University of Virginia in 1972. He has authored several technical papers and is a member of ASME, AIChe and Sigma Xi.

ABSTRACT

The Great Plains Gasification Plant in Beulah, North Dakota, converts lignite coal to pipeline quality synthetic natural gas (SNG) using high Btu technology. The rotating equipment for this grass root facility presented unique challenges, due to the process complexity and the size of the plant. The critical rotating equipment was sized and specified to maximize reliability and to minimize risks associated with extrapolation of field proven designs.

The plant rotating equipment was brought on stream ahead of schedule and continues to operate without any major problems. In spite of financial clouds over the project, the plant was a technical success and continues to operate with predicted onstream factors.

The equipment design audits were useful in identifying potential problems. Several problems encountered during the shop testing are discussed. These problems were resolved by working closely with the equipment vendors. The startup teams were organized early and were staffed with specialists from vendors and contractors to supplement the plant technical staff. Numerous startup problems were experienced and resolved expeditiously. These problems are briefly described along with observations for minimizing similar problems on future startups.

EQUIPMENT/PROCESS DESCRIPTION

The coal gasification plant in Beulah, North Dakota, is the first commercial-sized synthetic fuels project in the United States. It was designed to convert 14000 tons per day of North Dakota lignite coal into 137.5 million standard cubic feet per day (MMscfd) of pipeline quality synthetic natural gas (SNG). The project consisted of an open pit coal mine, gasification plant and an SNG pipeline. A simplified process unit arrangement schematic diagram for the plant is shown in Figure 1. A simplified material balance is depicted in Figure 2, and compressor train configuratio)size data are given in Table 1.

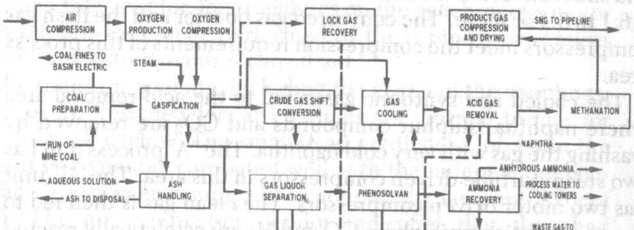

Figure 1. Process Unit Arrangement Schematic Diagram.

The plant design basis utilized two 50 percent capacity process units to keep the equipment sizes reasonable and to allow operation of at least one process unit, if the other unit was forced to shut down. A 2850 ton/day air separation plant supplies oxygen for the gasification process. It is among the largest plants of its type in the world. The "A" process unit has a steam turbine driven air compressor and a steam turbine driven oxygen compressor. In contrast, the "B" unit has synchronous motor drivers.

The gasification plant contains 14 (12 + 2 spares) Lurgi Mark IV dry bottom gasifiers. The coarse lignite is fed to these gasifiers operating at 430 psig along with steam and oxygen. The crude gas from the gasifiers is quenched to 370°F in waste heat

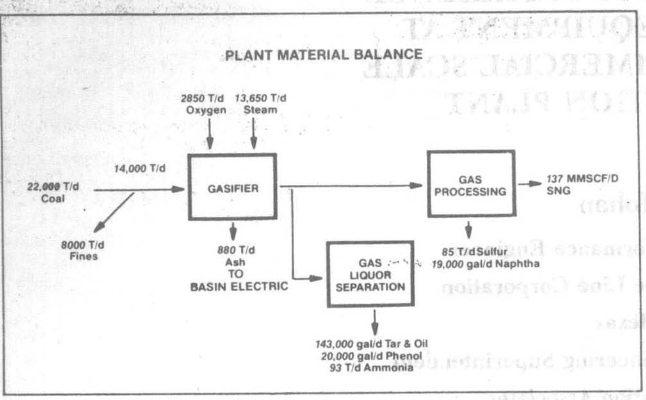

Figure 2. Simplified Plant Material Balance.

Table 1. Unspared Compressor Train Configurations.

Process Area	Service Description	Train ID	RATINGS Driver HP/	/Compressor RPM	Train Configuration
Oxygen	"A" Air Compr.	GB3001A	24233	4530	T -- CA
Oxygen	"B" Air Compr.	GB3001B	25000	4530	SM -- G -- CA
Oxygen	Oxygen Compr.	GB3004A	8650	11760	T -- C1 -- C2
Oxygen	Oxygen Compr.	GB3004B	9000	11240	M -- G -- C1 -- C2
Gasification	L.P. Lock Gas Compr.	GB1181	900	10147	M -- G -- C
Gas Cooling	Rectisol Flash Gas Compr.	GB1341	2000	15715	M -- G -- C1 -- C2
	Converted Gas Booster	GB1351	2620	7121	T -- C
Acidgas Removal	"A" Flash Gas Compr.	GB1401	3696	3200	T -- C
Acidgas Removal	"B" Flash Gas Compr.	GB1451	4000	3190	M -- G -- C
Acidgas Removal	"A" Ammonia Refrig. Compr.	GB1402	7935	8648	T -- C1 -- C2
Acidgas Removal	"B" Ammonia Refrig. Compr.	GB1452	8500	8648	SM -- G -- C1 -- C2
Methanation	"A" Methane Recycle Compr.	GB1701	7788	4335	T -- C
Methanation	"B" Methane Recycle Compr.	GB1721	8500	4459	SM -- G -- C
Gas Compression	"A" Product Gas Compr.	GB1901	11495	12863	T -- C1 -- C2
Gas Compression	"B" Product Gas Compr.	GB1921	11495	12863	T -- C1 -- C2

M = Induction Motor SM = Synchronous Motor C, C_1, C_2. = Centifugal Compressors CA = Axial Compressor

exchangers. The low pressure lock gas compressor is located in this area. Before methanation, the raw synthetic gas is cooled and shift converted to increase the ratio of H_2/CO in the gas from 2.6:1 to 3:1 or more. The converted gas booster and the flash gas compressors meet the compression requirements of this process area.

The cooled raw synthetic gas is fed to the acid removal area where naphtha, sulphur compounds and CO_2 are removed by washing the gas with very cold naphtha. The "A" process unit has two steam turbine driven compressors in this area. The "B" unit has two motor driven compressors. The clean gas is then fed to the methanation unit where CO and H_2 are catalytically reacted to form CH_4 and H_2O, producing SNG with a heating value of 970 Btu/scf. The methanation heat of reaction is used to generate 1250 psig steam, which is then superheated for use in the turbine drivers. The methane recycle compressor has gas inlet temperatures of over 500°F and is the most critical service in the plant. The SNG gas is dried in a glycol dehydration unit and compressed in the product gas compressors for feeding into the pipeline. The overall plant layout is shown in Figure 3. A plant photograph is presented as Figure 4.

In summary, the critical rotating equipment in the gasification plant consists of 15 unspared compressor trains. The 50 percent process unit "A" has six steam turbine driven compressors and the other 50 percent unit "B" has five motor driven compressors and one steam turbine driven unit. Any compressor trip cuts the total plant production in half. The remaining three compressor trains are not as critical, due to reduced production impact. The complexity of these trains varies considerably. The power ranges from 900 hp to 25000 hp, while the speed varies from 3190 cpm to 15715 cpm. In addition, the plant has over 20 partially spared compressors, fans, and blowers and over 600 process pumps.

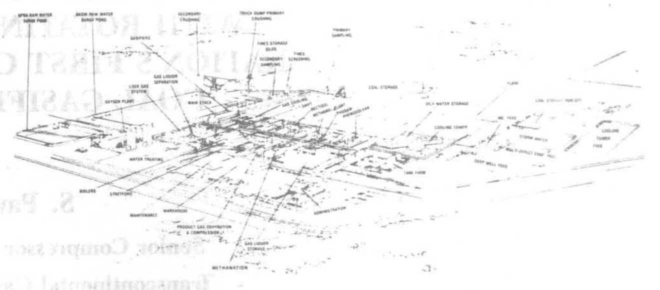

Figure 3. Plant Layout Schematic Diagram.

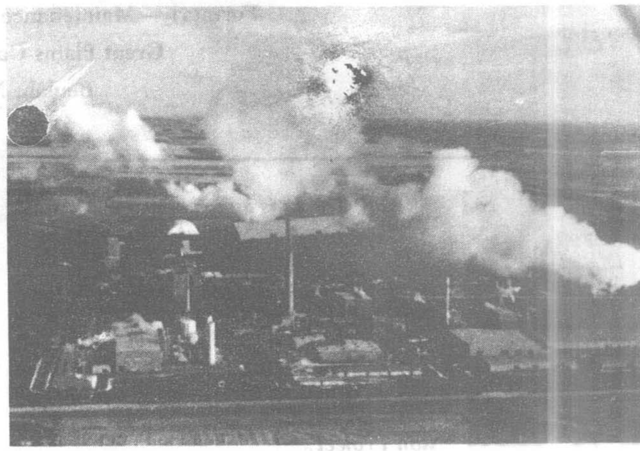

Figure 4. Plant Aerial View.

EQUIPMENT DESIGN BASIS

The "original" project team recognized that one of the key factors affecting the plant onstream factor would be the reliability and operability of the critical rotating equipment. To optimize these considerations, the following design guidelines were established:

- The critical compressor trains would be installed indoors, in heated buildings, to facilitate maintenance and construction during the severe winter season. Each building was to be equipped with a manually operated bridge crane.
- Generous spare parts including spare rotor, bearings, seals and labyrinths were to be purchased as part of the initial equipment order. A complete spare oxygen compressor was to be purchased.
- The maintenance shop would be equipped to handle normal compresssor repairs. Two balancing machines were to be purchased to handle the low speed balancing requirements of all the rotors.
- Each train would have a deck mounted compressor panel that would house the process and vibration instrumentation required to startup, operate and shutdown the equipment safely.
- Centrifugal compressors, gears, steam turbines and lube systems would be in accordance with API 617, 613, 612 and 614 specifications, respectively.
- Full mechanical tests per API 617 or API 612 specifications were to be performed on all rotors including the spare rotors.
- Performance tests were to be performed on the main rotor of each service, to minimize the chances of unseen performance deficiencies, especially for motor driven trains which would not have the speed flexibility of variable speed drivers.
- Dry membrane flexible couplings were preferred. The coupling hubs were to be fitted hydraulically to the shaft ends.

• Electronic governors were specified for turbine driven units. Redundant power supplies were specified for these governors. Turbine speed would have to be controlled from either the local panel or the control room.

• Each rotor was to have x-y proximity probes and a keyphaser probe to facilitate vibration monitoring. Each gear was to be provided with a casing accelerometer.

• Each tilting pad radial bearing was to be provided with two imbedded thermocouples to monitor the bearing condition.

• Each thrust bearing was to be of the self-leveling tilt pad design. Two thermocouples each were to be imbedded in both the active and the inactive sides of the bearing.

• The emergency shutdown systems were to be designed to allow tripping of compressor trains on either high radial vibrations or high axial movement. This was done to minimize catastrophic damage to the equipment, due to high vibrations.

Due to the limited engineering staff and a fairly tight startup schedule, it was decided to utilize the combined expertise of the engineering contractors, the equipment manufacturers and the outside consultants when practical.

DESIGN REVIEWS AND SHOP TESTING

The design reviews provided an opportunity to evaluate the adequacy of the equipment design prior to design completion by the vendors. Such reviews were conducted jointly with the engineering contractors and did indeed identify potential problem areas. Typically, the vendors were either requested to take corrective actions immediately or to develop alternate/backup designs to eliminate the problem, if it was observed on the test stand. Extensive shop testing was witnessed by representatives of the plant to assure compliance with API and project specifications. The following sections present the significant experiences through these particular project phases.

Methane Recycle Compressor

The methane recycle service required compression of methane, steam, and hydrogen during normal operation at inlet pressures and temperatures of 300+°F psig and 500+°F, respectively. However, during catalyst reduction, it was required to handle 100 percent hydrogen at similar pressures and temperatures. This severe service was considered a prototype design. Therefore, detailed design reviews and extensive testing was conducted.

A vertically split two-stage compressor with a conservative rotor bearing design and a proven impeller design was proposed by the compressor vendor. To eliminate the possibility of oil leakage into the process, a labyrinth seal design similar to the design used in an oxygen compressor was utilized. A seal skid was designed to control the flow of buffer gas and leakage to the recovery system under various operating conditions. A 120 hr mechanical/seal performance test at design pressures and temperatures was to be run on the vendor's test stand to validate this prototype design.

The 120 hr test run confirmed the adequacy of the overall design. In this closed loop test, difficulties were experienced in holding the loop pressure. Nitrogen had to be added continuously. The shaft vibration levels were about 0.5 mils to 0.6 mils and the bearing metal temperatures were in the range of 150°F to 170°F, even though gas temperatures varied from 400°F to 690°F. The critical speed was at 2400 cpm and somewhat higher than the predicted critical of 2300 cpm. The sealing system (Figure 5) and associated control hardware performed reasonably well. The control valves were in good operating ranges. However, sensing ports for the differential controllers were not in close enough proximity to the seals. To control seal leakage, a higher differential pressure across the sensing ports was necessary. The actual seal leakage rates were close to the design seal leakage rates.

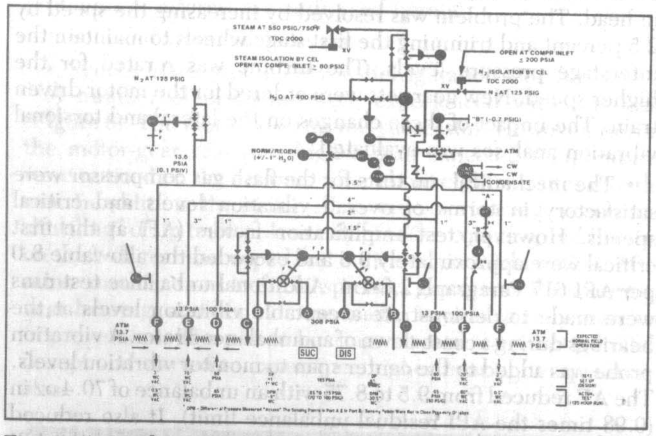

Figure 5. Methane Recycle Compressor Sealing System.

The casing was pressure tested at 350 psig with nitrogen gas after the previously described run. It did not pass the gas leakage test. The gasket between the head and the casing probably leaked during the 120 hr run. The differential expansion between the head and the casing caused the gasket to roll, resulting in leaks. The problem was solved by replacing the gasket with two fluorocarbon-silver plated metal 0-rings on a mirror finished surface. The unit received a helium leak test at 350 psig and a two day hot mechanical run. The unit was cooled down and pressure tested using nitrogen gas at 350 psig. No leaks were found. The inter-space between the two 0-rings was vented to the eductor on the buffer gas skid. A pressure switch was installed in this inter-space to warn of the O-ring failure.

Oxygen Plant Compressors

The oxygen plant was among the largest plants of this type and size in the world. One engineering contractor was awarded a turnkey contract, based on extensive experience. The steady state and transient torsional analyses on the synchronous motor driven trains were to be performed by the compressor vendors as well as a rotordynamics consultant.

The train components, including the gear and the couplings, were to be designed for seven times the normal torque for 10,000 load cycles. The analytical predictions of both the compressor vendors correlated well with the independent consultant. The oxygen compressor vendor used a damping factor of 0.03. The air compressor vendor indicated that they had extensive field experience with a damping factor of 0.04.

As a result of this analytical work, the rigid coupling between the gear and the oxygen compressor was modified to increase its peak torque carrying capability. On the air train, modifications were made to the gear and the coupling between the motor and the gear for the similar reasons.

All the oxygen plant equipment tested well during the shop tests, except for minor problems.

Other Services

The compressor trains in other services did not require significant extrapolation of equipment designs implemented currently in various petrochemical/refinery services. However, a number of problems of interest were experienced and are described briefly:

• For two services, the compressors did not meet the specified performance per ASME PTC-10 tests. These deficiencies were caused by smaller than design impeller tip widths and oversized fillet welds. In one case (GB 1181), the speed was increased by about three percent. New main and spare gear sets were procured by the compressor vendor. In the other case (GB 1402/2452), the second stage body was about eight percent low in head. The problem was resolved by increasing the speed by 2.5 percent and trimming the first stage wheels to maintain the interstage pressure levels. The turbine was rerated for the higher speed. New gear sets were ordered for the motor driven train. The impact of these changes on the lateral and torsional vibration analyses was evaluated.

• The mechanical run tests for the flash gas compressor were satisfactory in terms of overall vibration levels and critical speeds. However, test amplification factors (AF) at the first critical were approximately 9.5 and exceeded the allowable 8.0 per API 617 Paragraph 2.8.1.4. Additional unbalance test runs were made to demonstrate acceptable vibration levels at the bearings during a coast-down of an unbalanced rotor. A vibration probe was added to the center span to monitor vibration levels. The AF reduced from 9.5 to 8.75 with an unbalance of 70.4 oz in (0.98 times the API residual unbalance limit). It also reduced further to 7.75 with an unbalance of 174 oz in (2.41 times the API residual unbalance limit). The vibration level at the bearings with the later unbalance was about 0.7 mils. These tests demonstrated that the bearings would control vibrations reasonably. For this non-fouling service, the rotor was accepted without any modifications. The alarm and trip vibration levels were established after reviewing the modal deflections associated with the first critical speed.

• The calculated critical speeds and unbalance response results were reviewed for adequate separation margin from operating speeds per API specifications. During mechanical testing, the actual critical speeds were identified. For three services, the second critical speeds were close enough to the operating speeds to require unbalance sensitivity tests. A comparison of calculated and actual critical speeds, as well as amplification factors, is documented in Table 2. It was concluded that the "test stand" amplification factors are generally lower than the calculated amplification factors. This discrepancy could be partially explained by the assumption made in the rotor response calculations that the bearing stiffness and damping coefficients are determined by the static rotor loads.

Table 2. Comparison Summary—Calculated vs Actual Critical Speeds and Amplification Factors.

Compressor ID	Operating Speed	Predicted Critical Speed	Predicted Amplification Factor	Actual/Test Stand Critical Speed	Actual/Test Stand Amplification Factor
	RPM	RPM	-	RPM	
GB 1181 C	10147	3600	11.5	3650	6.5
GB 1341 C1	15715	6000	6.0	6820	4.0
C2	15715	7000	5.0	7676	3.2
GB 1351 C	7121	3800	2.0	4500	NM
GB 1401/51 C	3200	1525	22.0	1550	9.5
GB 1402/52 C1	8648	3180	9.0	3350	6.0
C2	8648	3550	9.4	3700	5.2
GB 1701/1721	4459	2300	6.2	2400	6.0
GB 1901/1921 C1	11495	5100	5.1	5100	4.0
C2	11495	4800	5.5	5000	3.2

NM - Not Measurable

Due to the shop test problems, the shipment of the compressors to the field was delayed. Testing was coordinated to minimize the impact on the startup schedule.

STARTUP ORGANIZATION

Due to problems encountered during the shop testing of rotating equipment, genuine concerns were expressed about the equipment reliability and its potential impact on the plant startup. To address these concerns and to develop a team for starting up rotating equipment successfully, the startup manager appointed a Compressor Task Force. This task force included representatives from the maintenance, design and operating groups and was initially requested to identify the potential problem areas and to develop an overall plan for equipment startup.

The task force identified the need to acquire diagnostic vibration equipment. The operating and maintenance groups were requested to get intimately involved in equipment turnover activities from the various contractors and subcontractors. A need for a compressor instrumentation engineer was justified to minimize anticipated problems associated with the lube and seal systems as well as the compressor panels. The task force meetings allowed the various personnel to develop a mutual understanding of each group's strengths and weaknesses. Various startup procedures including the initial commissioning checklists were developed by operating personnel for review by the design and maintenance engineering staff. Good working relationships and mutual trust were developed amongst the various groups. These intangibles were critically important in identifying and solving machinery problems later on during the startup.

The grassroots organization was carefully staffed in key positions with experienced individuals. But the majority of staff did not have relevant field/operating experience. As the startup progressed, the machinery startup engineering group was formally organized within the maintenance group to simplify coordination of numerous activities. The operators with machinery related interests were informally identified and were involved in solo and coupled runs of equipment. Classroom training was provided as necessary.

The startup organization was highly functional and worked together as a team to solve various vibration and operability problems associated with rotating equipment.

STARTUP ACTIVITIES

The project's startup period was scheduled to begin in mid-August 1983, and the plant was to have both trains feeding into the pipeline by December 1, 1984. The initial commissioning activities were fairly typical of any startup.

Lube Oil Systems

Great care was taken to clean up the lube oil systems and the associated piping. Problems experienced during the flushing of the lube and seal oil systems were similar to ones experienced by others [2] and by the author in an earlier startup. On one lube system, the long-runs of the field erected stainless steel piping were not cleaned adequately. This slowed the progress of flushing once the oil flushing through the bearings was started. During the first compressor run, a seal failure was experienced. Metallurgical analysis revealed entrapped sand, metal shavings, etc., in the failed parts. Additional flushing was undertaken and no seal problems have been experienced since then. The other eleven lube systems were flushed clean successfully.

The compressor instrumentation engineer was extremely helpful in setting up the lube and seal systems controls and in troubleshooting of system related problems [3]. Extensive "hands-on" training was provided to the operating and the instrument technicians.

Cold Alignment and Other Checks

At the time of equipment turnover from the engineering contractor, the final train alignments were witnessed by the maintenance personnel. "Soft-foot" problems were detected in many instances and were corrected before the equipment was formally turned over. Provisions were made to verify the initial growth estimates by using Essinger's long-stroke dial indicators [4].

Prior to solo runs, the bearings and seals were inspected for any damage, etc. Thrust floats were made for total casing travel and proper nozzle stand-off. Rotor float with the thrust bearings was also documented.

Solo Runs and Coupled Runs

Exhaustive lists of instrumentation settings around the compressor trains were developed by the machinery startup group in a format acceptable to the instrumentation technicians. This information, along with the special startup checklist prepared with the operating group, was extremely helpful in coordinating the activities necessary for successful runs. "Punch" lists were updated frequently to identify incomplete work items. They were also utilized as an effective training tool.

Vibration data were recorded and analyzed for each solo and coupled runs for reviewing and establishing baseline signatures. The quality and quantity of vibration data gathered depended on the severity of the encountered problem. Simple test reports documenting the results were issued promptly to communicate results and to assure that minor problems got corrected prior to the introduction of process gas into the compressors. Solutions to major problems were pursued aggressively, and are discussed later.

"Shake-Out" Period Activities

The probability of rotating equipment operation outside its safe operating envelope was considered highest during the initial shake-out period. The focus of activities was on keeping the plant online and with good reason, too. Recognizing this potential risk and the experience level of the operating staff, it was decided that a machinery engineer should be called out to the plant site, before a restart was attempted on a critical compressor train. A call-out list for each area of the plant was given to the operating superintendents. This approach helped in reducing the chances of restarting a mechanically distressed train. It was also useful in providing "hands-on" training to all the operating personnel and in refining the start-up checklists.

All the alarm and trip lists were reviewed and updated in light of the operating experience. The alarm and trip limits on axial position monitors were too tight and resulted in nuisance alarms. These limits were modified in a fashion similar to the recommendations of API 670, 2nd revision, 1985. The nuisance trips caused by the vibration/bearing temperature monitoring system malfunctions or by improper trip levels were minimal, if any. The functional tests of the anti-surge systems were also conducted. The opening times of the recycle valves were slow and were improved by adding boosters to the valve operators.

Informal reviews of the reliability and operability of the compressor trains were made periodically with the operating superintendents. The hot *vs* cold benchmark guage data were gathered and analyzed. Similarly, vibration data on the major compressor trains were gathered and analyzed for any unusual components. By April 1985, the maintenance engineers were keeping updated lists of "Shut Down Work Lists" for each compressor train in their areas. A preventative maintenance program for all the equipment including pumps, blowers, and fans was being implemented.

START-UP PROBLEMS

During this lengthy startup period, many problems were encountered that were resolved by persistence, knowledge, and team work with vendors and, to some extent, luck. Some of the problems of interest are addressed herein:

- Solo and Coupled Runs on
- Synchronous Motor Driven Refrigeration Compressor

A number of problems were experienced with this train (Figure 6). The motor solo run was successful. However, during the motor-gear solo run, higher than expected temperatures were noticed on the bearings after about eight minutes into the run. In addition, the pinion shaft was experiencing "precession shuttling" of up to 50 mils axially [5]. The coupling end bearing on the bull gear had wiped and indicated poor lubrication. After minor modifications to the oil spreader grooves on the spare bull gear bearing, another run was made. It yielded only marginal improvement. The pinion bearings were of the elliptical offset design and started to experience higher bearing temperatures also.

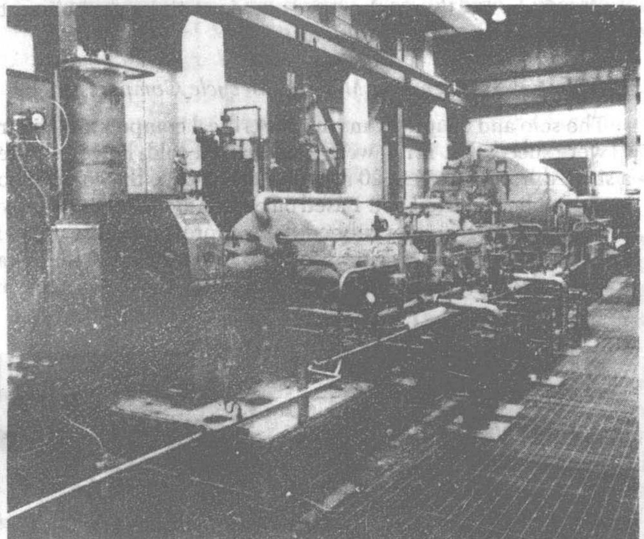

Figure 6. Synchronous Motor Driven Ammonia Refrigeration Compressor at 8500 hp and 8648 cpm.

The design of the bearings was reviewed with the gear manufacturer in light of this field experience. The bull gear bearing clearances were increased to values more in line with industry norms. The pinion bearing design was also changed to regular sleeve bearings with an anti-whirl pocket. This design was field proven and was predicted to have lower, but adequate stability characteristics, as compared to the elliptical bearing design. With these modifications, the motor-gear run was successful. Normal vibration and bearing temperatures were recorded. The pinion precession shuttling motion was reduced to less than 10 mils. To minimize potential startup delays, a backup tilt pad bearing design was developed in conjunction with a bearing manufacturer and manufactured on an expedited basis. Fortunately, the installation of these bearings was not necessary.

The compressor coupled runs with ammonia gas were conducted next. The normal audible gear tooth "clattering" as the synchronous motor accelerated through the torsional resonances was met with skeptism by people not familiar such equipment [6]. The comparison of actual and predicted acceleration times to the two torsional resonances, from the transient torsional analysis conducted by the compressor vendor, are depicted in Table 3. The compressor suction valves were not sufficiently automated and, consequently, the compressors would surge until the process was lined out.

In the interim, the difficulties in restarts after a process trip were experienced, due to normal difficulties in minimizing the

Table 3. Comparison of Calculated vs Actual Torsional Resonant Frequencies and Acceleration Times.

	Calculated Torsional Resonant Frequency (CPM)	Percent Of Motor Speed at which Resonance Occurs		Acceleration Times	
		Calculated	Actual	Calculated (Seconds)	Actual (Seconds)
Second Mode	1622	77.47	75.78	25.5	19.3
First Mode	929	87.11	86.94	28.5	22.5
Full Speed	----	-----	-----	31.5	28.5

process load for refrigeration services. The importance of starting the train with minimal process load was communicated to the operating group. The controls on compressor suction valves were upgraded to keep them closed until the motor was synchronized and then to open them to minimize surging. The vibration monitors were modified to double the alarm/trip levels for the duration of time the motor took to reach synchronous speed. A very specific and operator-friendly startup procedure was written by the author and an operating supervisor, to minimize these restart problems.

Hot Restart Problems on Methane Recycle Compressor

The solo and coupled runs on this critical compressor (Figure 7) were successful. It ran well when started cold; however, after a short duration of 15 to 20 minutes after a trip, the compressor could not be restarted hot. Based on carbon smoke coming out of the side vents, the rotor rubbed the compressor end seals, as it tried to go through the critical speed. Due to elevated temperatures and differential cooling of the rotor, the rotor took an excessive bow.

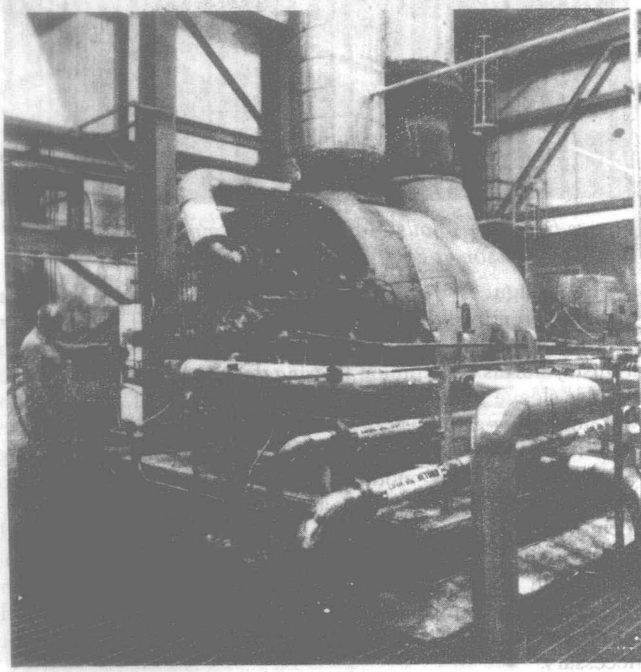

Figure 7. Synchronous Motor Driven Methanation Recycle Compressor at 8500 hp and 4459 cpm.

It was concluded that a turning gear was necessary for successful hot restarts and was to be procured on an expedited basis. The turning gear design was established in conjunction with the equipment vendors. The motor vendor recommended that a speed of 35 cpm would establish a minimum hydrodynamic oil film in the bearing. The train breakaway torque was determined by estimating the force required to turn a chain wrench. It was decided to install a turning gear driven by a 15 hp motor at the pinion blind end. The gear was to be engaged manually and was to disengage automatically as the motor came up to speed. This turning gear was manufactured and was ready for installation within ten weeks. It was installed and tested to show that it performed as designed.

In the interim, the shaft was to be manually turned 180 degrees every five minutes with a chain wrench, then the compressor could be restarted without problems. The operators were advised to check the variation in the gap voltages of the proximity probe prior to pushing the "start" button. In hindsight, a turning gear should have been procured for this application during the design phase.

CONCLUSIONS AND OBSERVATIONS

The plant rotating equipment was successfully commissioned ahead of schedule. This grassroots facility continues to operate reliably with predicted onstream factors. Although the team effort of plant personnel was critically important to the overall success, the following items were important in assuring equipment reliability and operability:

- The rotating equipment was sized and specified to maximize reliability and to minimize risks associated with extrapolation of field proven designs. The equipment was required to meet the applicable project and API specifications.
- Unique and difficult services were identified during the initial design phase. Additional design effort, reviews, and testing were planned to minimize unforeseen problems. Design audits were conducted jointly with engineering contractors to identify and correct potential problems. It was felt that the quality of vendor design effort was enhanced due to these audits.
- The shop testing was witnessed extensively to assure that all the design and quality control standards were met. The vendors cooperated willingly to correct all the major deficiencies.
- Prior to startup, a compressor task force of the operating, design and the maintenance personnel was formed. The task force was to address the reliability and operability concerns. Also, it was requested to formulate an equipment startup plan. Most of the recommendations were accepted by the management.
- Startup problems were recognized and expeditiously resolved with excellent cooperation from the vendors.
- The restarts during the "shake-out" period were monitored closely. The various alarm and trip levels were reviewed in light of the operating experience and were modified, if necessary. Operating problems were analyzed and addressed.

It was felt that the plant startup experience was very challenging and professionally enjoyable, in spite of difficult demands on time and family.

For future projects, the following observations are made:

- Conservative equipment selection is the key to minimizing equipment problems. If extrapolation of existing designs is necessary, additional design reviews and testing should be undertaken.
- Design audits including independent evaluation of the predicted critical speeds and amplification factors are highly recommended. It is money well spent and seems to improve the quality of the vendor design effort.
- Due consideration for a turning gear should be given for compressors with elevated suction temperatures, especially for motor driven units.
- For synchronous motor driven trains, the users should try to realistically estimate the process load imposed on the compressor for restarts. Sensitivity of the train torsional analysis and

restart times to additional process load should be evaluated as part of design audits.

- If minimal process load is assumed, additional instrumentation and controls including functional anti-surge systems should be specified during the initial design. The equipment specialist should make sure that these auxiliary systems are audited jointly with a control specialist as part of normal design audit.
- Early planning and good people are keys to a successful startup.

APPENDIX

Brief Project History

The Great Plains gasification plant in Beulah, North Dakota is the first commercial-sized synthetic fuels project in the United States. Utilizing two 50 percent capacity units, it was designed to convert 14000 tons per day of North Dakota lignite coal into 137.5 million standard cubic feet per day (MMscfd) of pipeline quality synthetic natural gas. The project consisted of an open pit coal mine, gasification plant and an SNG pipeline.

The process design work was started in 1973. Using Lurgi's coal gasification process, unique process designs were developed in conjunction with the South African Coal, Oil and Gas Corporation (Sasol) and other engineering firms [4]. Due to various environmental, financial and regulatory problems, the project was delayed for several years. Finally, in August 1981, the U.S. Department of Energy approved a conditional loan guarantee of $2.02 billion for the project. To support a late 1984 startup date, the equipment purchase orders were placed shortly thereafter. The construction activities were started immediately.

The nucleus of the startup team was assembled by August 1982. The startup activities formally started in August 1983. On July 28, 1984, for the first time, synthetic natural gas (SNG) from lignite coal was compressed into an interstate pipeline. The second train also started to produce SNG by December 1984. By February 1985, the SNG production rates were as high as 116 MMscfd, i.e., 88 percent of the design capacity was achieved and the plant production averaged about 70 percent of design [5]. During March 1986, the highest daily SNG production was 112.5 percent of the design rate. The monthly plant production averaged about 106.6 percent of design [6]. The amount of SNG pumped into the pipeline since the plant startup is depicted in Figure A.1.

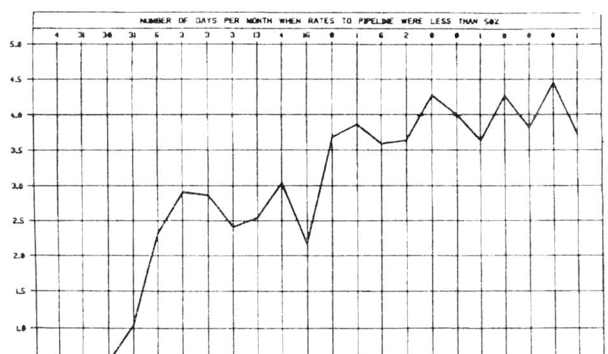

Figure A.1. SNG Passed Into the Pipeline.

Despite clouds over the project's financial viability due to falling energy prices, the project came in under budget and ahead of schedule. As of June 1986, the plant continues to operate with "design" on stream factors and production rates.

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ON THE MANUFACTURE OF IMPELLERS FOR TURBOCOMPRESSORS

by

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ABSTRACT

The efficiency of compressor impellers, apart from the design, depends on some features linked with their manufacture. These features are the accuracy of geometry, the surface quality obtained, and the blockages which are inevitable with covered impellers and are attributable to the joining method applied, e.g., greater blade thicknesses in the case of riveting, or narrowed cross sections, due to weld seams, in the case of welding. As a result of the ever increasing importance of efficiency, efforts have been made to further improve the manufacturing process.

The advantages and disadvantages of the various forming methods for high-quality impellers, such as milling with a relatively high accuracy, and involving high manufacturing outlay, are discussed. The various casting processes with a variety of possibilities to fulfill the requirements made with regard to the dimensional tolerances, process-dependent blade thicknesses and manual work for achieving certain specified accuracies are described.

The efficiency of impellers with closed passageways, that is, with a cover or shroud disc, is superior to that of impellers with open passageways. This means that when no cast impellers are used, suitable methods must be available for joining the blades and the cover. For maximum accuracy and minimum blockage, the high-vacuum brazing method has become more and more accepted during the past few years. However, the manufacture of impellers applying the brazing method becomes problematic for large diameters. In addition to vacuum brazing, manual and machine welding processes continue to be applied, including slot welding for two-dimensionally curved blades.

Diffusion welding and electron-beam welding have not found acceptance since high costs, unfavorable crevices, and sharp edges, along with possible deformations, have a negative influence on the behavior of such impellers during operation.

It is true that the sophisticated riveting process of some manufacturers for two-dimensionally curved blades requires great wall thicknesses; but, it is also true that high-accuracy impellers can be produced by this method. In addition, the vibratory behavior of such impellers is favorable due to system-inherent dampening. Moreover, riveted impellers will tolerate high stresses when the Bauschinger effect is taken into account.

The manufacture of highly stressed impellers for turbocompressors is based on the availability of suitable materials and adequate test methods.

INTRODUCTION

The impeller is the most essential part of a radial-flow turbocompressor. In view of the ever increasing importance of its performance, the geometrical design of an impeller is governed by the laws of aerodynamics, thermodynamics, and stress. Therefore, there is only a small choice of compromises to the effect of altering of the geometrical data with the aim of adopting a specific manufacturing process. As a consequence, special methods for the manufacture of turbocompressor impellers had to be developed or already known methods had to be adjusted to satisfy the changing demands made of a turbocompressor impeller.

These manufacturing methods, the criteria for their selection, and the resulting quality assurance measures are described herein. The emphasis, however, is put on process compressors of specific design and manufacture to meet customers' requirements. Series-manufactured compressors will not be addressed.

The field of application and the configuration of three-dimensional (3D) geometrical turbocompressor impellers with various types of the three-dimensionally curved blades is shown in Figure 1 as a function of the impeller specific flow coefficient

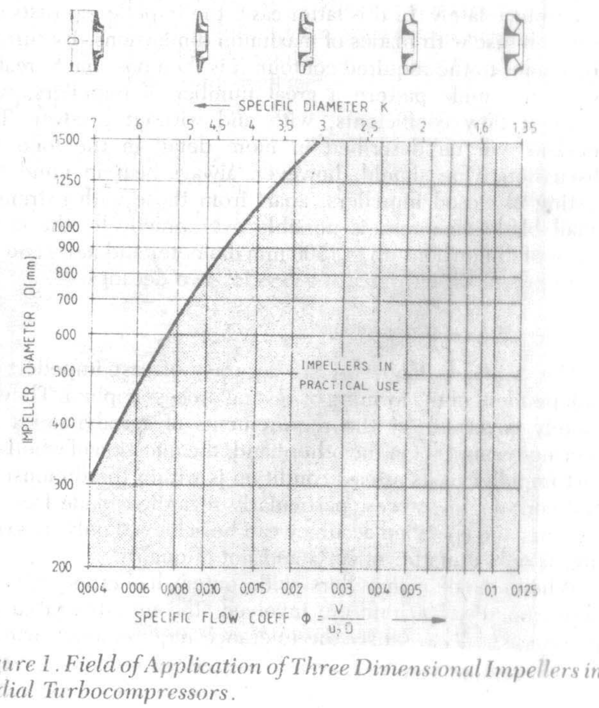

Figure 1. Field of Application of Three Dimensional Impellers in Radial Turbocompressors.

and of the impeller diameter. This diagram represents the latest impeller designs used in practice. Nevertheless, there still exists a wide range of two-dimensional impellers. Therefore, such impellers will also be discussed.

The manufacture of impellers involves complex technological problems. There are several options to systematically deal with these problems and they will be discussed as follows:

The manufacture of the hub disc with the blades will be discussed first, since manufacturing the blades and hub disc from one piece has become common. Next, special importance will be attached to the question of joining the cover to the blades. Certain criteria for the selection of materials and for the heat treatment of the blanks or of the completed impellers emerge from the variety of optional manufacturing methods. The methods and the extent of quality assurance will be governed by the manufacturing process selected and by the special requirements of materials to be satisfied.

The outlining of technical details regarding the sequence of manufacturing operations then results in the criteria to be selected for a specific case. When making this selection, one must also bear in mind that the great number of technical options is to be reduced to such an extent that the continuous application of a given manufacturing process in a workshop is ensured, taking into account the spectrum of impellers to be manufactured. Only then can the know-how and practice in working with a given process be kept on the highest standard possible.

THE MANUFACTURE OF HUB DISCS WITH BLADES

There are two principle methods of manufacturing the blank of the hub disc, i.e., casting and forging. Compared to closed impellers made from forgings, with the cover always being a separately manufactured part (except where the channels are manufactured using the electro-erosion process which is applied relatively seldom), the cover can be integrally cast when producing closed impellers by casting. This, however, implies that adjusting the impeller geometry to the requirements of thermodynamics necessitates a costly modification of the pattern. Compared to the impeller with a fixed contour of the cover that has been in use for quite some time, the principle of the impeller with a variable contour of the cover has been gaining in importance lately. In this latter case, the impeller consists of a cast hub disc with blades of maximum dimensions. By turning the blades to the required contour, it is then possible to realize with one single pattern a great number of impellers, with different flow coefficients, with and without a cover. This method will be described in more detail in the following discussion. One should, however, always bear in mind that casting of closed impellers, apart from those with extremely small blade channels, is possible over practically the entire range of dimensions up to 1500 mm diameter and more and has been practiced with great success for two decades.

Casting of Impellers

The demands made on the accuracy of cast impellers are independent of the forming or casting process applied. They are merely governed by the requirements of aerodynamics and thermodynamics. On the other hand, the question of whether a cast impeller in its as-cast condition is within the dimensional and contour tolerances, particularly regarding blade faces, or whether the specified accuracy can be achieved only by grinding, is only a matter of costs and not of quality.

Where smaller impellers with outer diameters of up to approximately 500 mm are involved, the investment casting process is able to satisfy the tolerance requirements, without necessitating any rework. For larger diameters of up to 1500 mm and larger, it is common practice to use a single ceramic core containing all the blade faces involved (monobloc). The formerly used sand core is inadequate for producting sufficiently accurate impellers.

These two methods will be briefly described in the following subsections. In the further course of discussion, the method of casting closed impellers is taken as being similar, as mentioned before, if the monobloc process is applied. Narrow passage closed impellers manufactured using the investment casting process have not been produced with convincing results. Thus, closed impellers of smaller diameter also have been produced using the conventional forming process, where this is possible under the aspect of channel size. In addition to the lack of flexibility in designing impellers, the casting of impellers with covers features another drawback: channel side faults in the cover, in the hub disc and in the blades which are difficult to find and repair, depending on their size and their accessibility.

Investment Casting Method

The basis of casting impellers following the investment casting process is a pattern of low-melting wax, as shown in Figure 2. With regard to its dimension, this wax pattern is identical to the final casting produced in that it must consider all shrinking effects during solidification of the steel casting and all changes to the contour which may occur on cooling or on heat treatment.

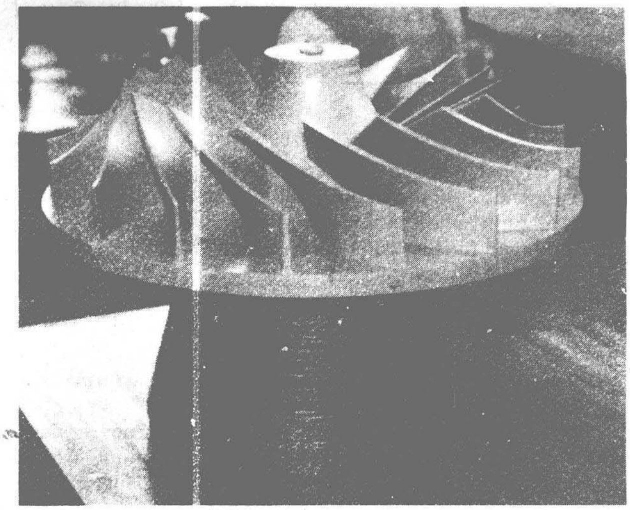

Figure 2. Wax Pattern for Investment Casting.

This wax pattern is normally made in a high precision steel mold. The surfaces of this mold, which are especially important because of the blade geometry, are made of aluminium on a five-axis milling machine. They are used for making cores of low-melting metals, which represent the channels in the mold that is shown in Figure 3. The pattern must be designed so that opening the mold and forcing out the part is possible after pressing in and solidification of the wax.

Specially selected ceramic masses of different grain sizes and composition are now applied layerwise, with the aid of manipulators which are usually computer controlled to ensure a uniform layer thickness (Figure 4). During the burning of this ceramic mass, the wax melts out, thus creating the mold for the turbocompressor impellers. Casting takes place at such high ceramic temperatures that even thin cross-sections flow out well. Allowances on the blade surfaces are, therefore, not necessary. Apart from occasional straightening operations, the

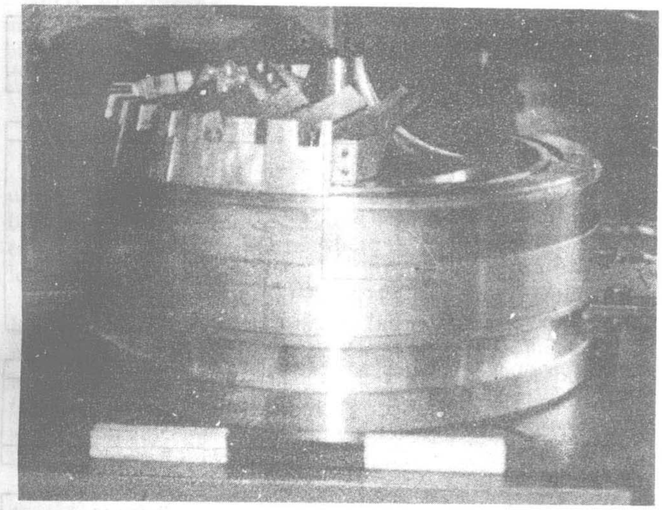

Figure 3. Die from Steel for Producing Wax Patterns.

Figure 4. Covering of Wax Pattern with Ceramic on a Computer Controlled Manipulator.

blades and the hub disc need no furher machining in the channel area after casting.

Casting with Monobloc Cores

For larger impellers, it is no longer possible to make wax patterns of one single piece. Wax patterns assembled from individual segments or a core of solid ceramic must contain all of the impeller channel, including the blade surfaces. The method, using wax patterns from individual segments, has not been a general success, since improving of the accuracy is not possible, in contrast to the monobloc core method.

A condition for making a monobloc core is blades of a material that melts out or gasifies on burning of the ceramic material. Consequently, these blades, shown in Figure 5, must be made of wax or of plastics with a process that is very similar to the investment casting process.

The blades are inserted into a pattern of wood or of plastics, and are located in their exact position in such a manner that a highly accurate ceramic core of one single piece (monobloc) is obtained, as shown in Figure 6.

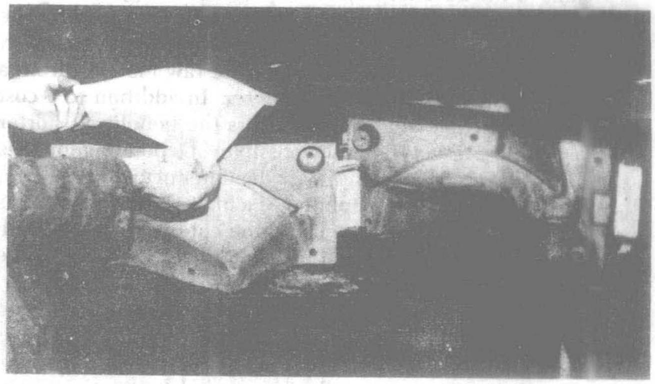

Figure 5. Blade Made from Wax and Mold Used.

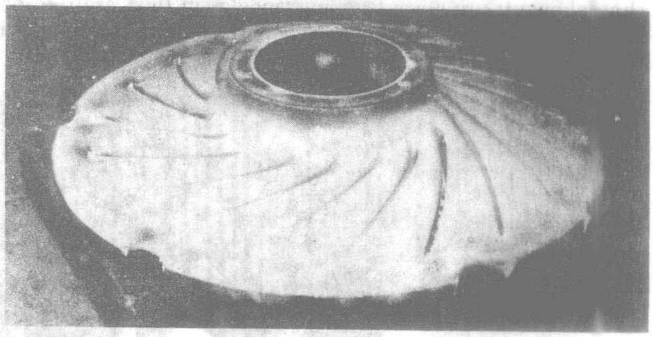

Figure 6. Monobloc Core for Two Dimensional Impeller with Outer Diameters over 500 mm.

The sequence of casting is not any different from conventional casting. Since the mold, however, is relatively cold before casting, and does not flow out so well, as in the case of the investment casting process, it is necessary to make allowance, especially on the thin blade inlet edges. In addition, the effect on the casting surface cannot be ruled out, because of the extended influence of the atmosphere and of the mold materials during solidification and cooling of large impellers. Impellers cast with a monobloc core will thus obtain their final dimensions only with 100 percent grinding of all faces not machined on the subsequent turning. The accuracy thus obtained, however, is, relative to the size of the castings, equal to that of small impellers made using the investment casting process. Some cast impellers in the as-cast condition and in the ready-for-delivery condition are shown in Figure 7.

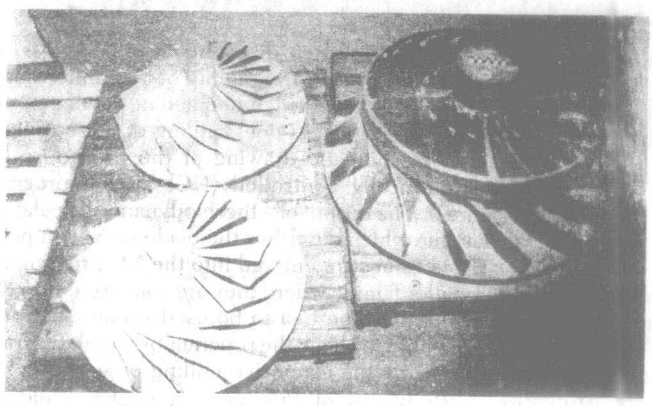

Figure 7. Three Dimensional Impellers in the As-Cast Condition and Ready for Final Machining.