

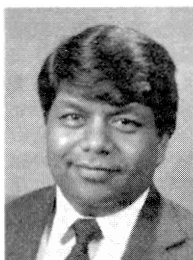
TESTING AND STARTUP EXPERIENCE WITH ROTATING EQUIPMENT AT NATION'S FIRST COMMERCIAL SCALE COAL GASIFICATION PLANT

by

S. Paul Mohan

Senior Compressor Performance Engineer
Transcontinental Gas Pipe Line Corporation
Houston, Texas

Formerly—Maintenance Engineering Superintendent
Great Plains Gasification Associates
Beulah, North Dakota



Paul Mohan is a member of the Transmission Services Staff at Transcontinental Gas Pipeline Company, Houston, Texas. He is involved in projects aimed at improving reliability, operability, performance and maintenance of the pipeline compressor stations. In 1982, he was assigned to the rotating equipment startup team at the Great Plains Gasification Project.

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 Dakotalignite 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 configuration/size data are given in Table 1.

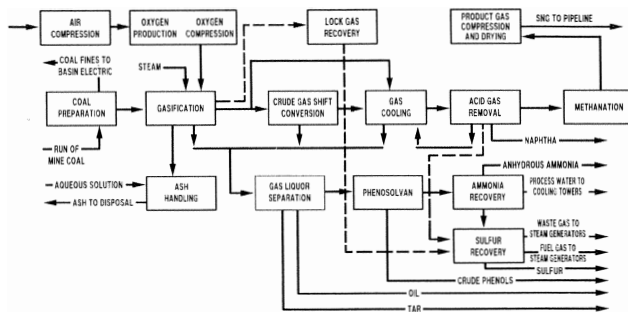


Figure 1. Process Unit Arrangement Schematic Diagram.

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

- 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/back-up 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 neces-

sary. The actual seal leakage rates were close to the design seal leakage rates.

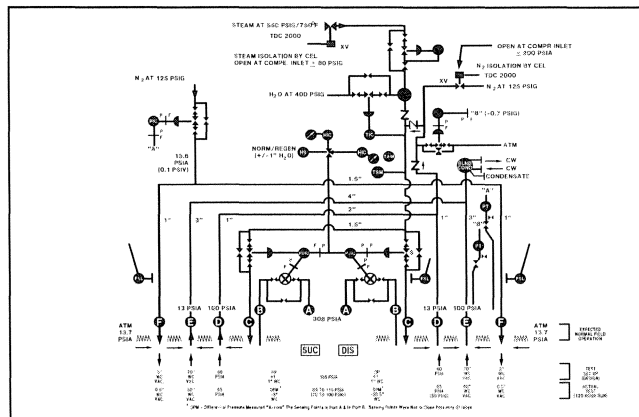


Figure 5. Methane Recycle Compressor Sealing System.

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 O-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 O-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		Actual/Test Stand	
		Critical Speed	Amplification Factor	Critical Speed	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.

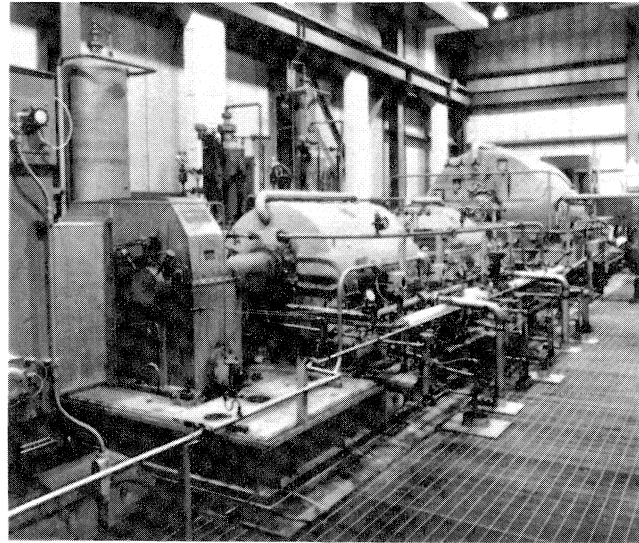


Figure 6. Synchronous Motor Driven Ammonia Refrigeration Compressor at 8500 hp and 8648 cpm.

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 skepticism 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 (CFM)	Percent Of Motor Speed at Which Resonance Occurs		Acceleration Times	
		Calculated	Actual	Calculated (Seconds)	Actual (Seconds)
Second Mode	1522	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.

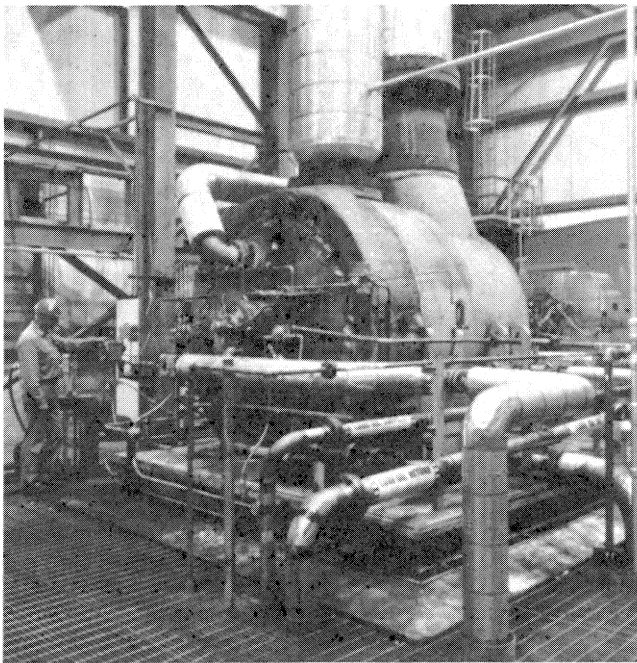


Figure 7. Synchronous Motor Driven Methanation Recycle Compressor at 8500 hp and 4459 cpm.

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.

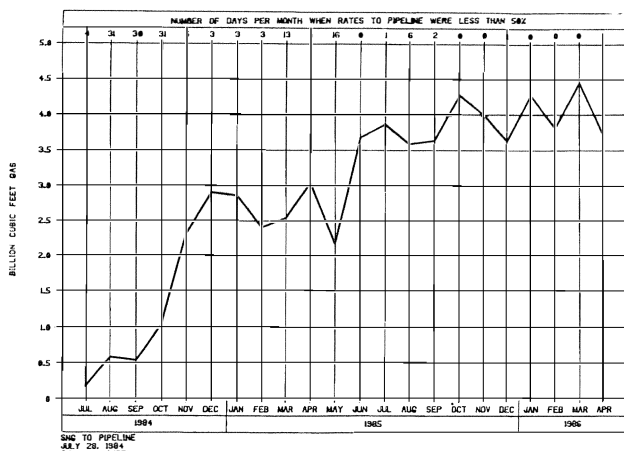


Figure A.1. SNG Passed Into the Pipeline.

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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