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DEVELOPMENT OF TURBINE DRIVEN CENTRIFUGAL COMPRESSORS FOR NON-CONDENSIBLE GAS REMOVAL AT GEOTHERMAL POWER PLANTS

FINAL REPORT
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DISCLAIMER

Abstract

Initial field tests have been completed for a Non-Condensible Gas (NCG) turbocompressor for geothermal power plants. It provides alternate technology to steam-jet ejectors and liquid-ring vacuum pumps that are currently used for NCG removal. It incorporates a number of innovative design features to enhance reliability, reduce steam consumption and reduce O&M costs. During initial field tests, the turbocompressor has been on-line for more than 4500 hours as a third stage compressor at The Geysers Unit 11 Power Plant. Test data indicates its overall efficiency is about 25% higher than a liquid-ring vacuum pump, and 250% higher than a steam-jet ejector when operating with compressor inlet pressures of 12.2 in-Hga and flow rates over 20,000 lbm/hr.

Introduction

This project has been funded by Barber-Nichols Inc., Pacific Gas and Electric Co., UNOCAL Corp., and the U. S. Department of Energy (Grant DE-FG07-95ID13391). The turbocompressor was designed, manufactured and tested by Barber-Nichols Inc. Pacific Gas and Electric Co. performed all the installation work and provided design, test and development support. UNOCAL Corp. provided design support.

The goal of this program was to develop and demonstrate a high efficiency, reliable, cost effective turbocompressor to remove NCG from the condensers at geothermal steam power plants. In geothermal plants, the gas produced from the resource is a mixture of steam and NCG. After this mixture is expanded through the main turbine, the NCG must be removed from the condenser and compressed to ambient pressure before it can be properly abated.

Most plants currently use steam-jet ejectors, sometimes in combination with liquid-ring vacuum pumps, for NCG removal. In some plants, up to 20% of the steam produced from the resource is required by the ejectors for NCG removal. This is steam that would otherwise be available to produce power. This program has developed a turbocompressor for NCG removal that significantly reduces the parasitic steam flow requirement.

Assembly and installation drawings for the turbocompressor are shown in Figures 1 and 2. It consists of a single-stage, axial flow steam turbine that drives a single-stage centrifugal compressor. The turbocompressor design incorporates several innovative features to reduce costs, enhance reliability, and provide other operating advantages.

- It requires much less parasitic steam (or comparable parasitic power) than other technologies.

Figure 1

NCG TURBINE-COMPRESSOR ASSEMBLY

Barber Nichols
Denver, Colorado USA

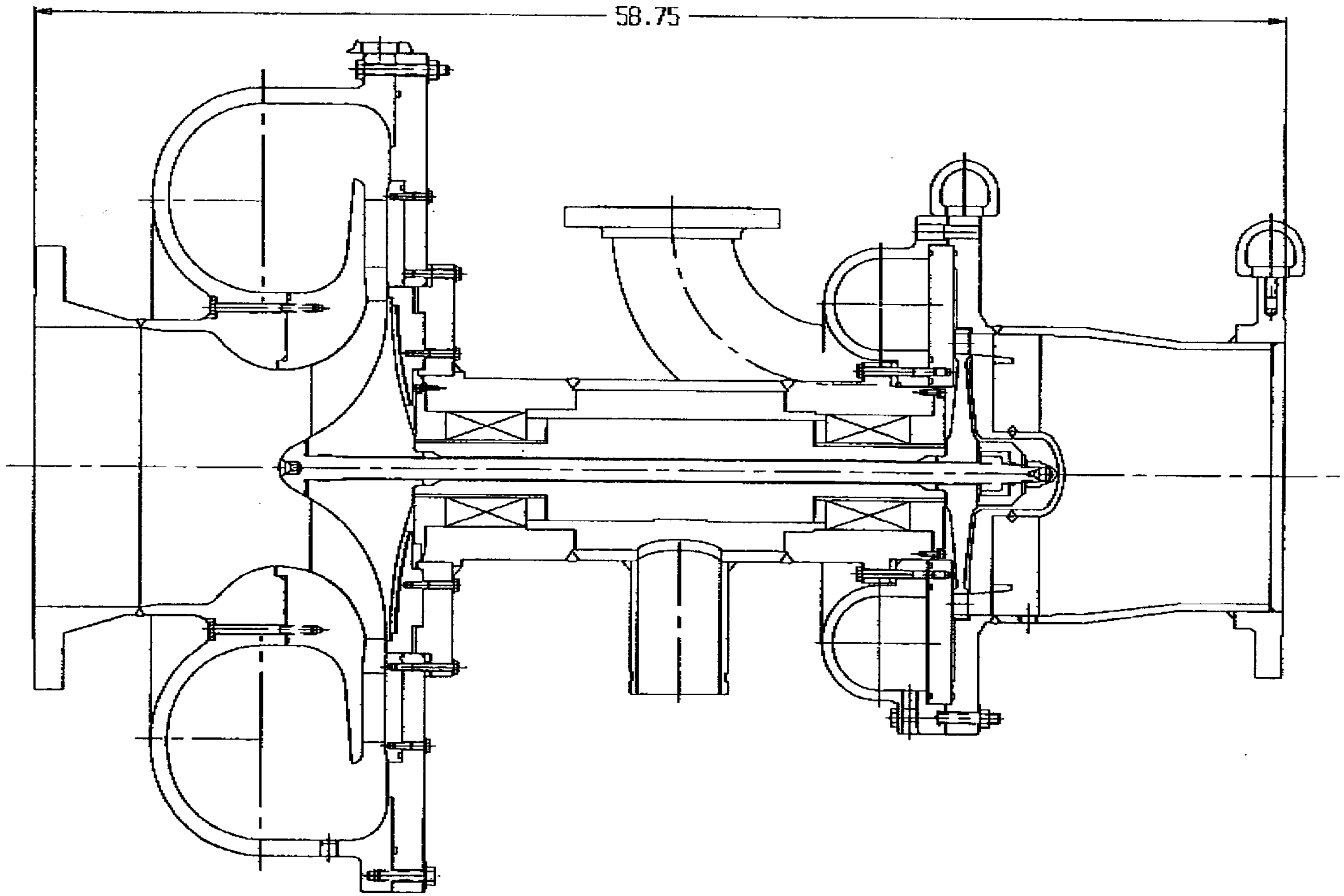


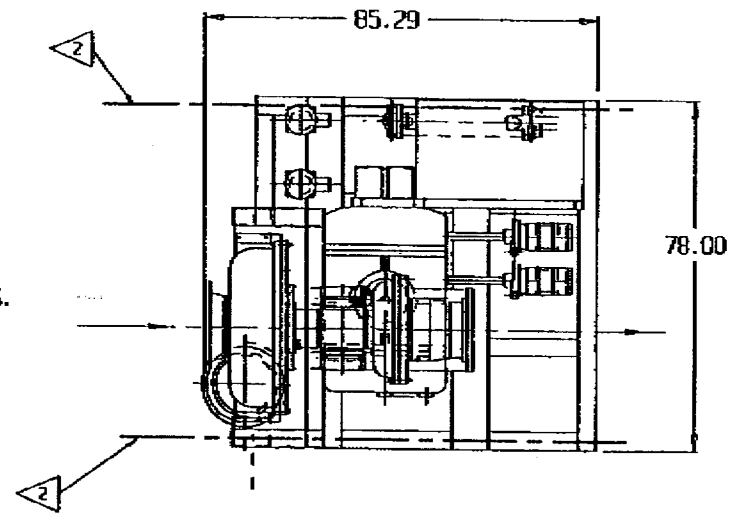
Figure 2

NCG TURBINE-COMPRESSOR INSTALLATION

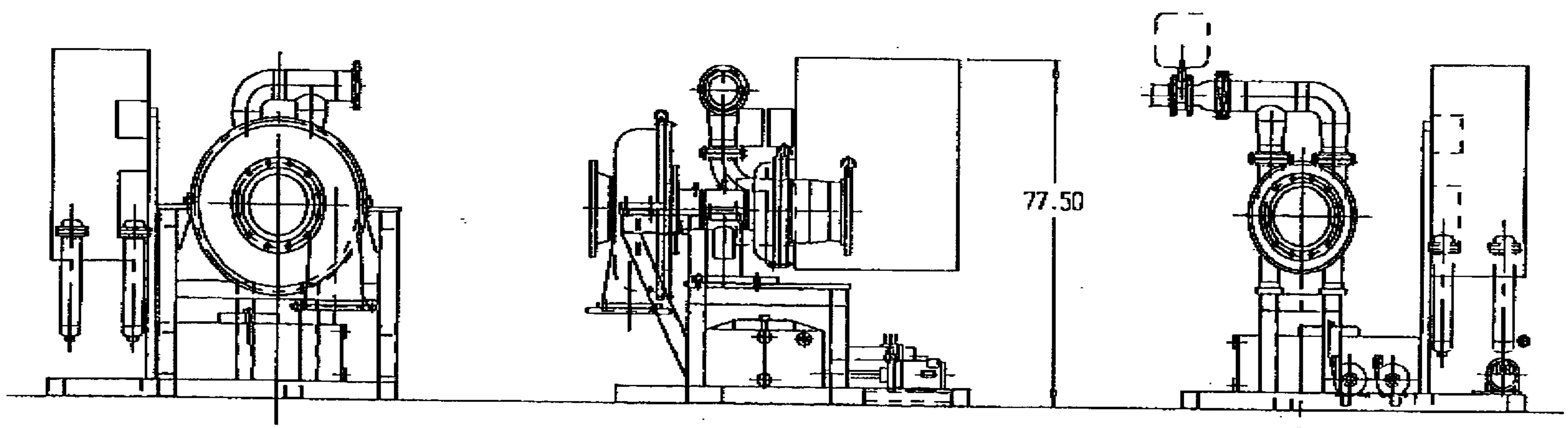
Barber Nichols
Denver, Colorado USA

NOTES;

1. ESTIMATED WEIGHT 5900 LBS.
2. SUPPORT SKID ON NOTED CENTERLINES.



PLAN



- With minor adjustments to the trim of the flow components, it can handle changes in steam pressure and NCG flow rates as the resource matures.

- The rotating assembly is supported on water lubricated hydrodynamic bearings. This eliminates the need for an oil lubrication system and shaft seals. Since shaft seals are typically the high maintenance component in high speed equipment, this significantly enhances the reliability of the turbocompressor.

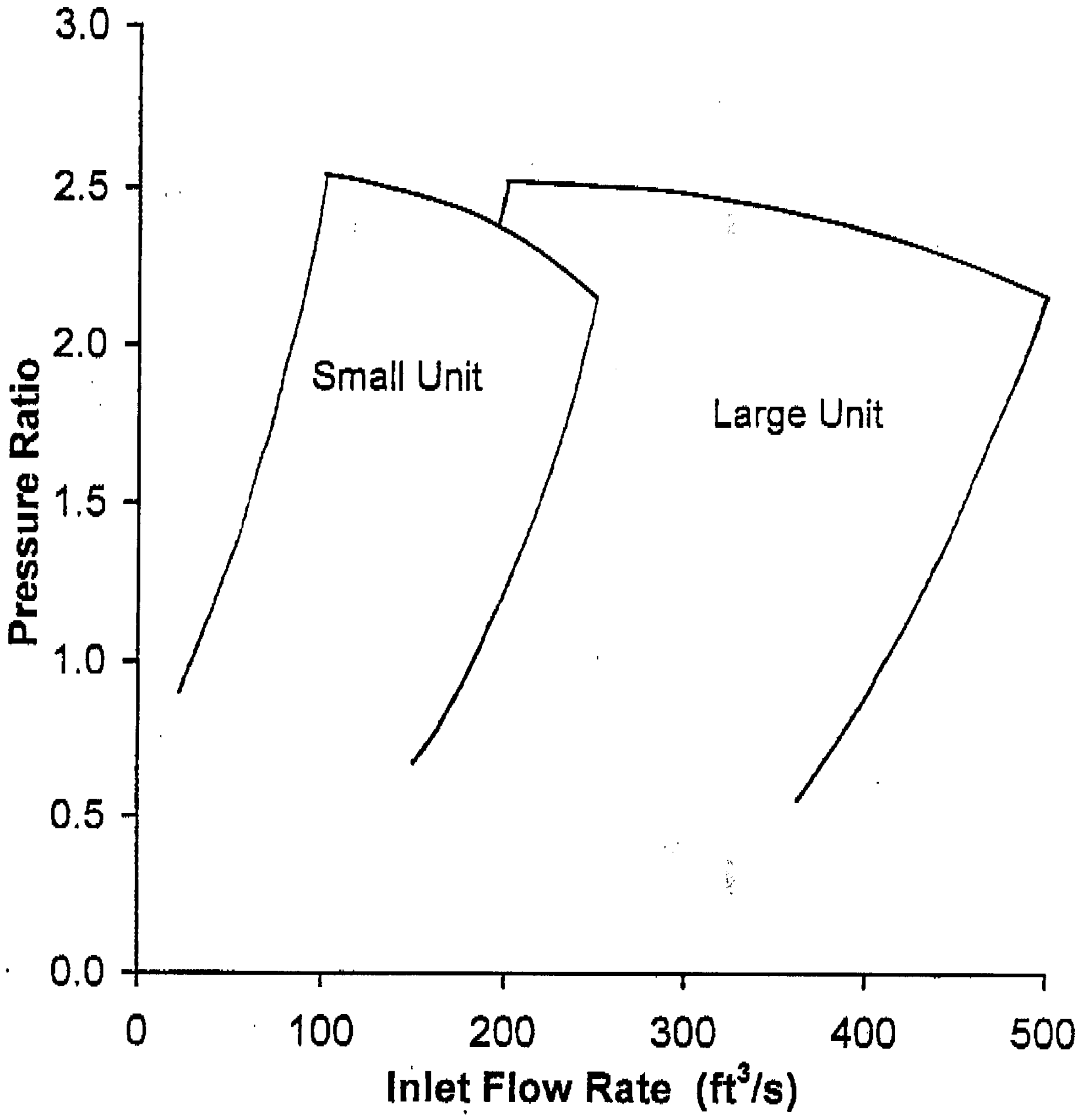
- It can handle larger inlet volume flow rates and lower inlet pressures than liquid-ring pumps. Consequently, it can be used for the lower pressure stages in an NCG removal system.

- It is well suited to retrofit applications since it is a small, stand-alone machine that can be mounted on an existing turbine deck. For many retrofit applications, the pay back period is less than a year.

The scope of the current project was to develop and demonstrate a turbocompressor for the third stage of the NCG removal system at The Geysers Unit 11 Plant (stages one and two are steam-jet ejectors). In order to insure that the turbocompressor will be useful for a wide range of different plant conditions, a design study was conducted. The investigation considered the NCG removal requirements for a number of plants that covered a wide range of conditions: sizes from 5 - 100 MWe, dry steam and flash plants, condenser pressures from 1 - 4 in-Hg, and NCG flow rates from 2,000 - 40,000 lbm/hr.

In order to cover this range of conditions, two basic sizes of turbocompressors are required. A small compressor, that will handle the lower flow rate requirements for the third stage at Unit 11, and a large compressor, that will handle higher inlet volumetric flow rates. A map of the operating envelopes for the small and large compressors are shown in Figure 3. By adjusting the trim of the turbine and compressor components, small and large units can be combined to cover the range of requirements discussed above.

Figure 3. Compressor Operating Envelopes



The process capabilities of the small turbocompressor are as follows.

Compressor (Envelope shown in Figure 3)

Maximum Inlet Flow Rate	250 ft^3/s	
Maximum Pressure Ratio	2.5	

Turbine

Maximum Inlet Pressure	120 psig	
Maximum Exit Pressure	10 psig	

Combined

Design Point Efficiency	50%	With no entrained liquid at the compressor inlet

The bearings, bearing housing, and shaft that have been developed for the Unit 11 turbocompressor are designed so that they will meet the requirements for both the small and large turbocompressors. The turbine and compressor components (turbine plenum, nozzle and rotor and compressor impeller, diffuser, and volute) that have been developed for Unit 11 are designed so that they will cover the entire flow envelope shown in Figure 3 for the small compressor. The turbine and compressor components for the large unit will be developed after more field experience is accumulated on the small unit.

LABORATORY OPERATING EXPERIENCE

Initial testing of the turbocompressor was done at Barber-Nichols test facilities during June - August 1996. The following tests were completed.

Functional checks of all safety controls: overspeed, lube minimum pressure, lube maximum temperature, shaft maximum vibration, shaft axial travel limit, and bearing maximum temperature.

Multiple tear down inspections to check all parts, and in particular, the water lubricated bearings.

Operated the turbocompressor for 25 hours and 50 start-stop cycles.

Collected compressor and turbine performance data (discussed in the next section).

The only significant development issue that came up during laboratory testing was material selection for the water lubricated bearings and the adjacent running surfaces on the shaft. Initial materials stood up well during continuous operation. However, they would only handle a limited number of start-stop cycles, where the fluid film breaks down and adjacent surfaces rub. The materials that were finally selected have superior surface lubricity characteristics and handled 50 start-stop cycles (roughly 5 years of plant start-stop cycles) without any indication of wear.

FIELD OPERATING EXPERIENCE

In August, 1996 the turbocompressor was shipped to The Geysers. The scheduled outage for Unit 11 was delayed until late November and the installation was completed in January, 1997. The turbocompressor system passed all plant safety and trip tests, and the plant was brought on-line with the turbocompressor in late January.

During initial operations at the plant, several development problems came up and were successfully corrected. The bearing journal materials had to be changed. Hardened stainless steel journal materials worked well during laboratory testing, but did not provide an adequate margin against Stress Corrosion Cracking (SCC). During initial field tests, the journal sleeves were damaged by SCC and the turbocompressor tripped on high vibration. The sleeves were replaced with a proprietary hardened material that is much less prone to SCC and there have been no further problems with them in 4500 hours of operation.

Modifications were required to slinger components that are located between the bearings and the turbine and compressor wheels. The slingers keep the bearing lube water from escaping into the turbine and compressor process streams. They were also designed to limit contact between the lube water and the process streams in order to minimize absorption of process gasses in the lube water. However, as evidenced by the SCC damage to the shaft sleeves, process gasses were being absorbed in the lube water. The slingers were modified to improve the isolation between process streams and the lube water. Since modifications have been completed, there have been no indications of significant migration of the process gasses into the lube water.

The controls were modified to handle an unanticipated transient condition that was encountered during shut down. After a plant trip, there is reverse flow through the compressor as ambient air backflows through the NCG line into the condenser. This reverse flow causes the shaft to spin backwards. A valve was installed at the compressor discharge to eliminate the backflow.

These modifications were completed February - April 1997. From April to August, Unit 11 was on-line with the turbocompressor operating. During this period, the turbocompressor operated approximately 2500 hours over a wide range of conditions. Towards the end of the operating period, bearing temperature and lube water pressure readings indicated that lube water passages were becoming obstructed. A disassembly inspection revealed that calcium carbonate was scaling the lube water passages and restricting the bearing water flow.

The calcium carbonate comes from the hard make-up water that is used for the lube water system. The turbocompressor uses a closed-loop water lube system with very little blow-down. In this operating mode, the lube system concentrates the hard make-up water to the point that calcium carbonate precipitates. Operating procedures have been modified to use a reverse osmosis water treatment and adjust blow-down to limit lube water Total Dissolved Solids (TDS) to potable water levels. The turbocompressor has been in operation for approximately 1500 hours since the lube system was modified and there are no indications of scaling in the lube passages.

To date, the turbocompressor has been on-line at Unit 11 for over 4500 hours. As discussed above, only minor modifications have been required to address the development problems that have arisen. During this field test period, all important aspects of the mechanical design of the unit have been validated.

- Conservative design criteria were used for all turbocompressor components in order to maintain maximum stresses well below SCC thresholds. Aside from journal sleeves (that were subsequently changed to a different material), there has been no indication of SCC in any turbocompressor components, in spite of the very aggressive Geysers SCC environment.

- Rotordynamic characteristics of the turbocompressor are well behaved with no observable critical speeds in the operating range. Shaft vibrations are well within normal limits for this class of machine with typical vibration levels on the order of one mil of radial eccentricity.

- There have been no indications of vibration or fatigue problems in any turbocompressor components.

- In spite of high flow rates of entrained liquid at the compressor inlet (discussed in the following sections), there are no indications of significant erosion damage.

- The simple PLC control system performs all automatic start-up, shut-down and fault monitoring functions and interfaces properly with the plant control system.

As long as adequate water flow is maintained, there have been no problems with the water lubricated journal and thrust bearings. They have stiffness and damping characteristics that are well suited for this class of machine, there has been no measurable wear, and materials have been identified that are compatible with the operating environment and have good inherent lubricity characteristics for start-stop cycles.

LABORATORY PERFORMANCE TESTS

The turbocompressor was first tested at Barber-Nichols test facilities. These tests were conducted with the compressor acting on a dry air stream that was throttled upstream from the compressor to produce vacuum inlet conditions. The turbine was driven with saturated steam from a facility boiler. Due to limitations with the facility boiler, it was necessary to operate the turbine far off-design during laboratory tests. It was only possible to obtain performance data at speeds up to 11,000 rpm (65% of the 17,000 rpm design speed). It was also necessary to operate the turbine with a restricted arc of admission and at pressure ratios over 8.0 (more than twice the 3.5 design pressure ratio).

Due to the large difference between field operating conditions and laboratory test conditions, there was no attempt to use this data to predict performance at field operating conditions. However, since these were the only tests that could be conducted with laboratory instrumentation, there is a high level of confidence in the data and they have been useful in terms of validating Barber-Nichols performance prediction methodology, and evaluating some conflicting data from field tests.

Performance data from laboratory tests at 11,000 rpm are shown in Figures 4 and 5. This data indicates that at the design compressor inlet flow rate, the pressure ratio actually produced by the compressor is about 3% below the design prediction. At this same flow rate, the overall efficiency (combined turbine, compressor and mechanical efficiency) measured during the air tests agrees very well with the design prediction. It should be noted that overall efficiencies for these laboratory air tests are low because the turbine is operating at extreme off-design conditions. The predicted overall efficiency with the turbine and compressor at design conditions is 0.54.

Figure 4. Laboratory Test Data

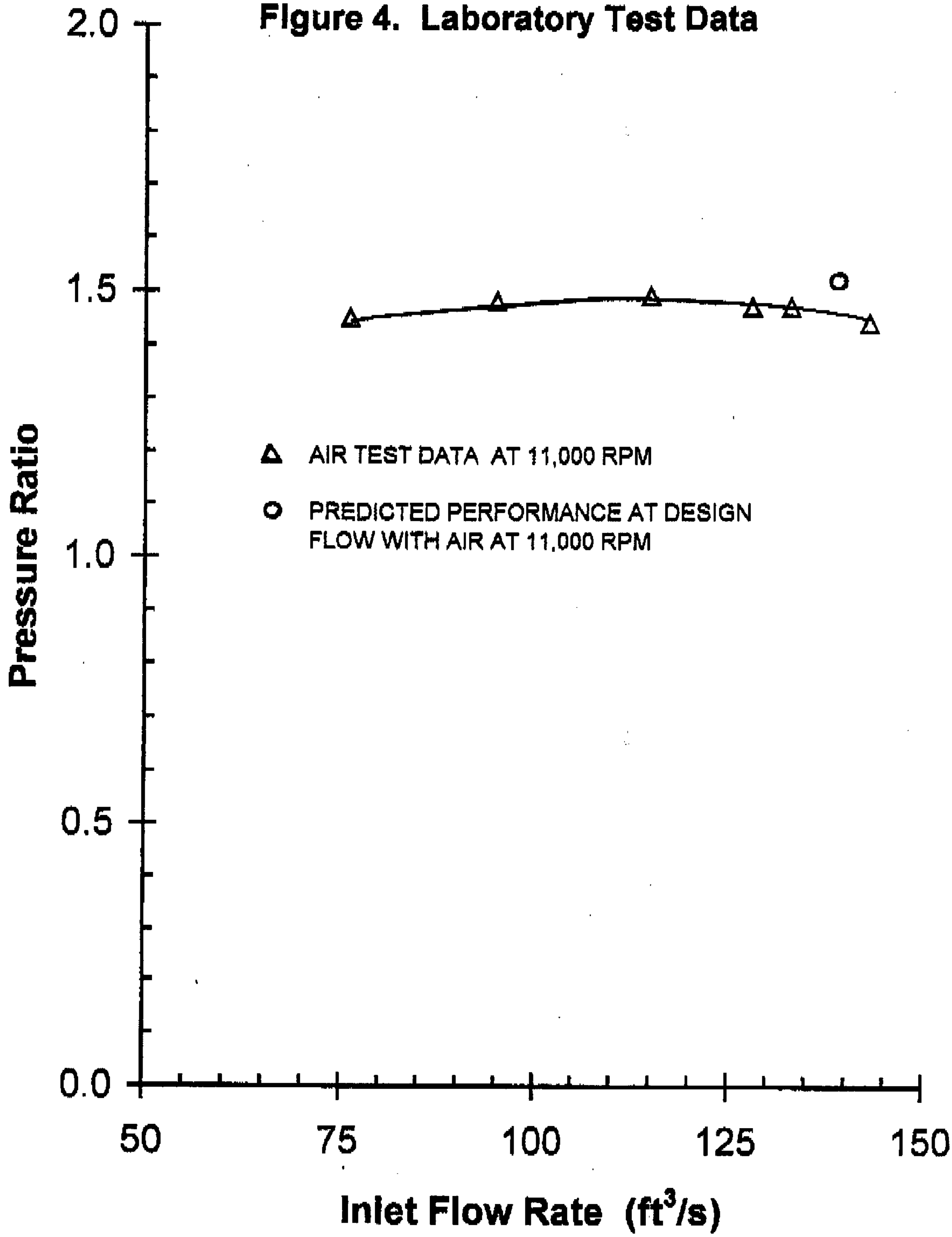
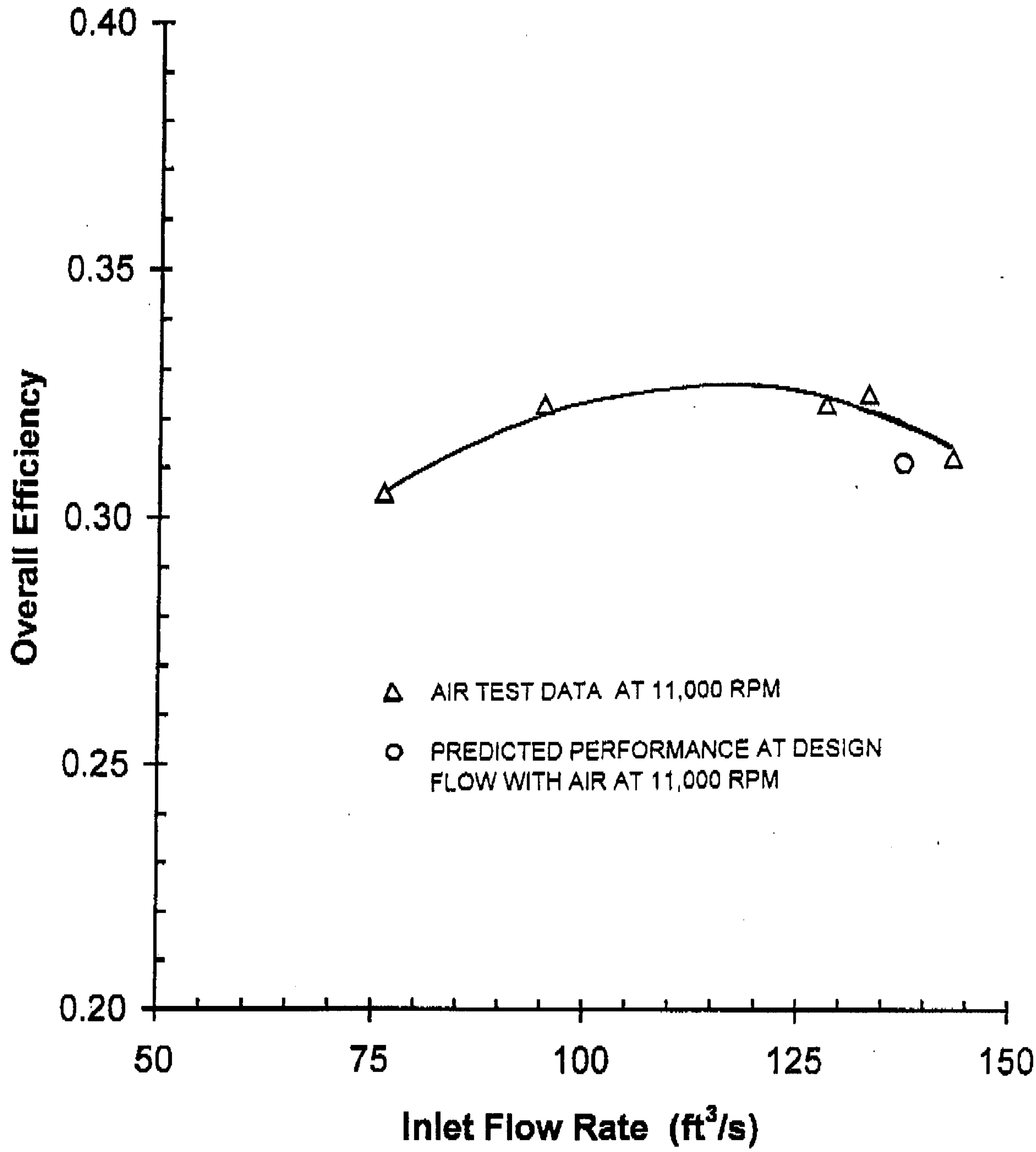


Figure 5. Laboratory Test Data



NOTE: PREDICTED EFFICIENCY WITH COMPRESSOR AND TURBINE AT DESIGN CONDITIONS IS 0.54

Figure 6. Field Test Data

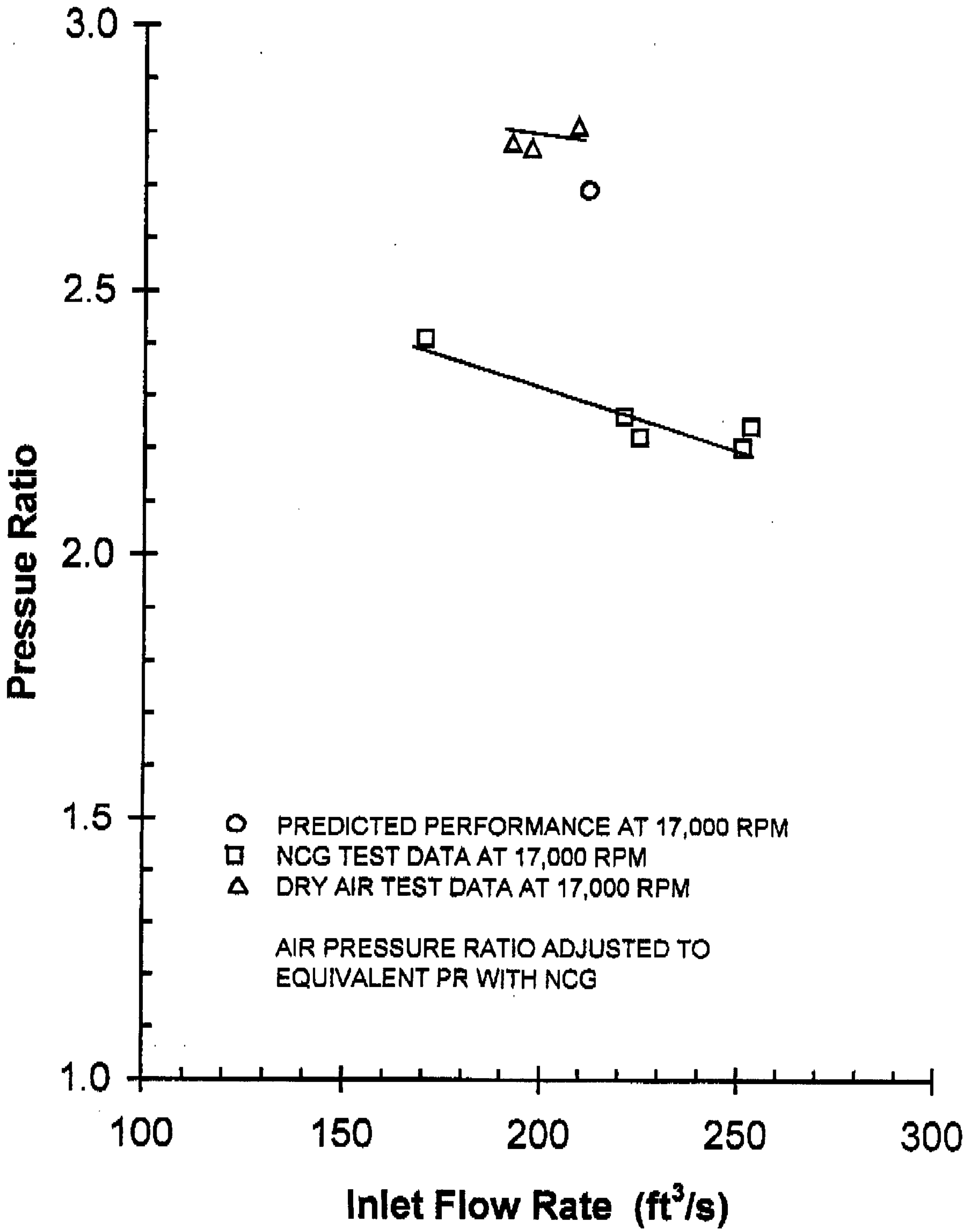
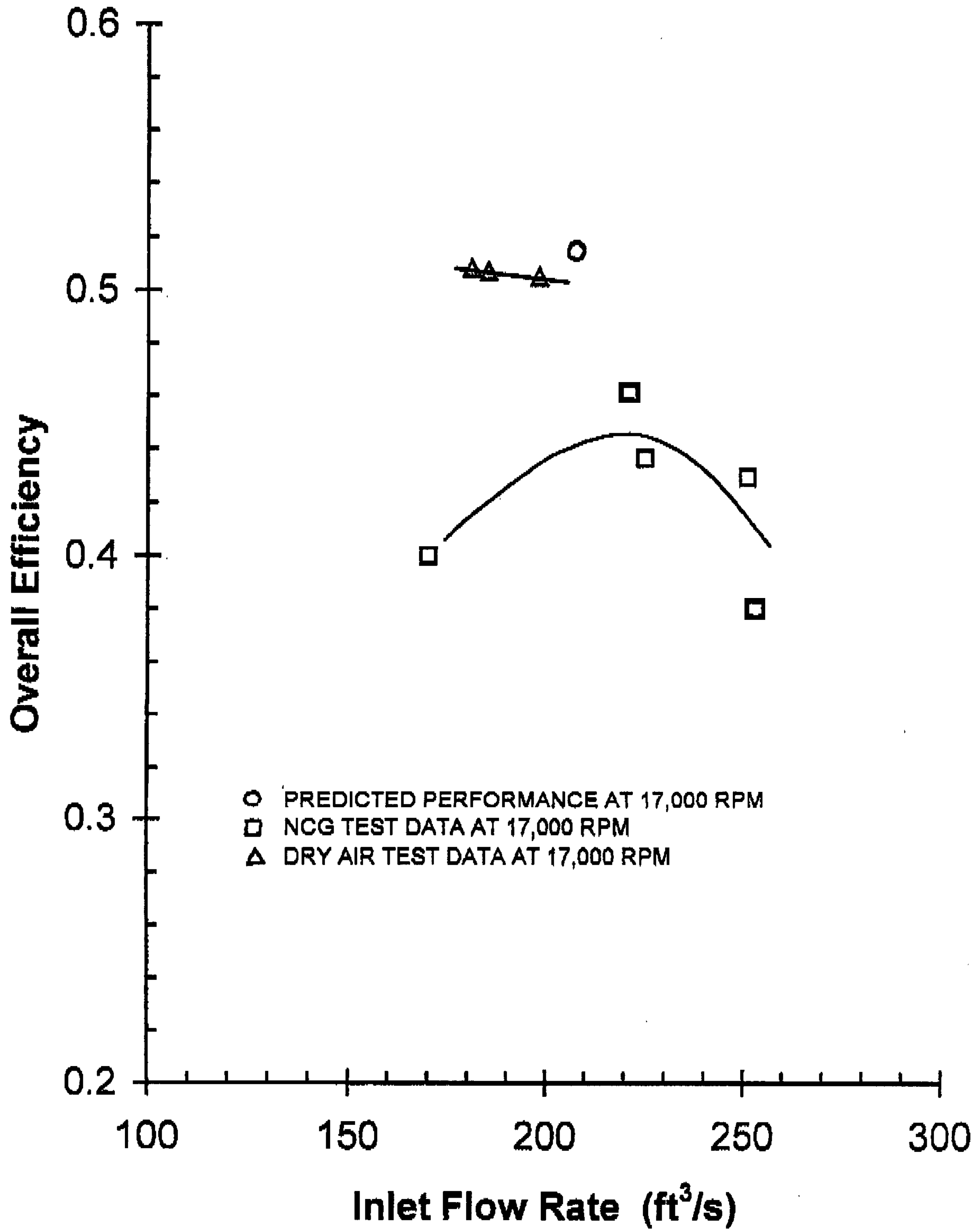


Figure 7. Field Test Data



The amount of entrained liquid can be estimated as follows. At design speed, the measured temperature rise across the compressor is approximately 40° F. Based on performance predictions, and on dry air test data, the temperature rise without entrained liquid should be approximately 140° F. The difference between the predicted and measured temperature rise is due to vaporization of entrained liquid water. The minimum water flow rate that could account for the temperature difference is a mass fraction of approximately 2.7% liquid water in the NCG stream. This liquid fraction would account for the cooling if all of the liquid was vaporized in the compressor. However, it should be noted that the liquid fraction could also be significantly higher than this value if the liquid is not completely vaporized in the compressor. With instrumentation available at the plant, it is not possible to determine the entrained liquid fraction more precisely.

To estimate how much of the measured turbocompressor performance is due to entrained liquid in the NCG stream, a set of field tests were run with dry air flowing through the compressor. These test results are also shown in Figures 6 and 7. In order to compare NCG and dry air test results, the pressure ratio for the air data has been adjusted to the equivalent pressure ratio for NCG. (Differences in specific heat and molecular weight result in different pressure ratios across a compressor.)

Figures 6 and 7 show that the dry air data falls very close to the predicted pressure ratio and efficiency points. This suggests that most of the performance shortfall is due to entrained liquid. Modifications are being made to the inter-condenser to reduce the amount of entrained liquid. The effectiveness of these modifications will be evaluated in future testing.

CONCLUSIONS

Field operations at The Geysers Unit 11 power plant have demonstrated that the turbocompressor is a rugged, reliable piece of equipment. Turbocompressor components show no signs of mechanical problems or corrosion attack after more than 4500 hours of operation in the aggressive Geysers steam and NCG environment. The use of water lubricated bearings has eliminated the need for a shaft seal, which is the generally the major reliability problem in high speed equipment. In the last 4000 hours of operation, the only problems encountered were due to scaling of the water lube passages. With recent system changes to improve the quality of the lube water, it is anticipated that the turbocompressor can operate for 12 months between scheduled maintenance.

Test results at Unit 11 suggest that there is a high fraction of liquid water that is entrained in the NCG stream flowing through the compressor. This entrained liquid penalizes the overall efficiency of the turbocompressor. However, even with the entrained liquid, the turbocompressor has an overall efficiency of about 45%. A liquid-ring vacuum pump and a steam-jet ejector for the same service have overall efficiencies of approximately 35% and 18% respectively. Testing with dry air suggests that the overall efficiency of the turbocompressor will be over 50% when the amount of entrained liquid is reduced.

The overall efficiencies for the turbocompressor, liquid-ring vacuum pump, and steam-jet ejector are inversely proportional to their steam flow requirement (if the steam expansion pressure ratios are the same). Compared to a steam-jet ejector, the steam conserved by the turbocompressor can produce an additional 1400 KWe of net electrical power by expanding it through the power turbine. If this energy is valued at $.03 per kw-hr, annual plant revenues are increased by $360,000 if when it is operating with the turbocompressor instead of a steam-jet ejector. For many retrofit applications, it is estimated that the payback period for a turbocompressor installation is less than a year.

The turbocompressor also provides other improvements to the overall plant steam rate. At Unit 11, the discharge steam from the turbocompressor is back-pressured to approximately 4 psig and a portion of it is used for main turbine seal steam. This conserves additional high pressure steam that would otherwise have to be throttled for seal steam. It should be noted that if the turbocompressor steam was expanded to main condenser pressure instead of 4 psig, only half the steam flow would be required and the increase in plant net power output would double.

Further testing will be conducted at Unit 11 to demonstrate reliability and further document performance improvements. However, test results to date indicate that the turbocompressor offers viable new technology that should be considered with steam-jet ejectors and liquid-ring vacuum pumps in identifying the most cost effective mix for a NCG compression system.