

# REAPPLICATION OF AN INDUCTION STEAM TURBINE TO EXTRACTION COGENERATION SERVICE

by

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tion instead of 40 psig induction. This project was developed and managed by plant engineering at the Green Lake plant. The development, reapplication, remanufacturing, and startup phases of the steam turbine generator are documented. Contributions from the OEM, after market suppliers, and independent consultants were vital to the project's success. Measures taken to ensure the performance of the contractors and confidence in the reliability of the end products are described.

## INTRODUCTION

This 15 megawatt extraction steam turbine supplements an existing 17 megawatt condensing steam turbine already in operation at the Green Lake, Texas chemical plant. The acrylonitrile process is an exothermic reaction of air, ammonia and propylene, which not only creates the main product, acrylonitrile, but also generates large quantities of high pressure steam. This steam drives equipment in the plant as well as the existing 17 megawatt turbine generator.

## NEW REQUIREMENTS

In March 1989, the plant underwent an expansion which produces an excess steam capacity of 170,000 lbs per hour of 650 psig, 750°F steam at the new maximum production rate. Approximately 80,000 lb/hr could be let down from 650 psig into the plant's 50 psig steam system, while the excess 90,000 lb/hr would had to have been vented. The existing water demineralization train would not be able to makeup enough high quality boiler feed to meet this added demand.

The scope of this project was to provide an operational controlled extraction turbine generator capable of condensing the excess steam as well as generating electricity for export into the utility grid. The project time frame was for startup within one year. Since original equipment manufacturers could not meet the delivery required by this schedule, the used equipment market was investigated.

## THE USED EQUIPMENT MARKET

The expansion project was originally proposed in outline form in late 1987. A condensing steam turbine with approximately 80,000 lb of extraction capability at 55 psig was determined to be the ideal equipment to maintain the steam/condensate system balance. At this time, preliminary inquiries were made with the original equipment manufacturers. None of them could meet the desired equipment requirements while adhering to the deliveries required by the schedule. The used equipment market was then reviewed more in terms of meeting the schedule than in terms of cost.

## ABSTRACT

Reapplication of used power equipment can shorten project schedule and decrease capital cost significantly. The following is a broad overview of the reapplication of an FCC airblower steam turbine to 15 megawatt cogeneration service in B.P. Chemicals, acrylonitrile plant in Green Lake, Texas. The turbine, an Elliott 2NV9 with 40 psig induction, was originally installed in a Louisiana refinery in 1965. A 1982 refinery revamp left this equipment mothballed in place until March 1988, when it was purchased by B.P. Chemicals. This turbine was then salvaged and modified to meet a new operating requirement, 55 psig extrac-

Literature from various used power equipment vendors along with after market manufacturers and after market repair shops were requested for information on the availability of steam turbines in the used market. Initial contacts just requested minimal data on the equipment and an approximate price. Virtually all the equipment encountered at this stage was World War II vintage or earlier and ranged in price from around \$400,000 to one million dollars. Based on this data, a scope of supply was developed and sent out to the various equipment vendors. The scope of supply requested the availability of a steam turbine generator and condenser with refurbishable auxiliaries having a nominal rating of 11 to 15 megawatts (MW) with steam conditions of approximately 650 psig inlet pressure, 750°F inlet temperature and a condensing pressure of two inches HgA. A synchronous generator was requested and, as an option, extraction capability was also requested. The company requested that the turbine be rated for outdoor service, in good condition, and manufactured after 1960. Several bids were received from all the vendors. Most of the offerings were for pre-World War II turbine generators. Four of the vendors all quoted a particular late model 20 MW turbine generator set. Quotes received for this one particular generator were one million, 1.5 million, two million, and three million dollars; which gives an idea of the margins expected by some used equipment vendors. The foregoing prices did not include refurbishment. Further inquiries into the "particular" turbine revealed the maze of legal constraints involved in acquiring this "particular" used piece of equipment from various vendors. It is recommended that engineers and project leaders work closely with an experienced purchasing specialist to avoid potential legal entanglements.

In the final analysis, the company chose a 20,000 hp induction-condensing steam turbine previously used to drive a refinery FCC blower (Figure 1). The turbine was manufactured by Elliott Company in 1965. It was designed to startup on 600 psig, 700°F high pressure steam and operate on 200,000 lb/hr of 40 psig steam induction to condensing. It was the only turbine offered which had more than 15,000 lbs/hr of capability for extraction/induction. The operating speed was 3300 rpm to 3650 rpm. It was built for a maximum throttle flow of 176,000 lb/hr of high pressure steam or a maximum induction flow of 200,000 lb/hr, 40 psig steam. A suitable synchronous generator (Figure 2) was also located as a match to the steam turbine. The 15 MW synchronous generator was originally driven by a gas turbine in a Gulf Coast chemical plant. Previously salvaged, it was immediately available for refurbishment, however, it had no exciter.

The company purchased the steam turbine (Figure 3), condenser, all desired auxiliaries, spare parts and the generator for under \$300,000. An additional contract was let for the demolition and removal of the steam turbine and auxiliaries from the Louisiana refinery. The removal cost just over \$100,000. The

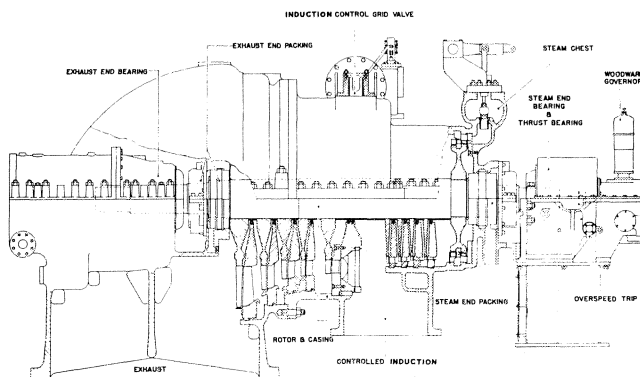


Figure 1. Sectional View of Turbine.

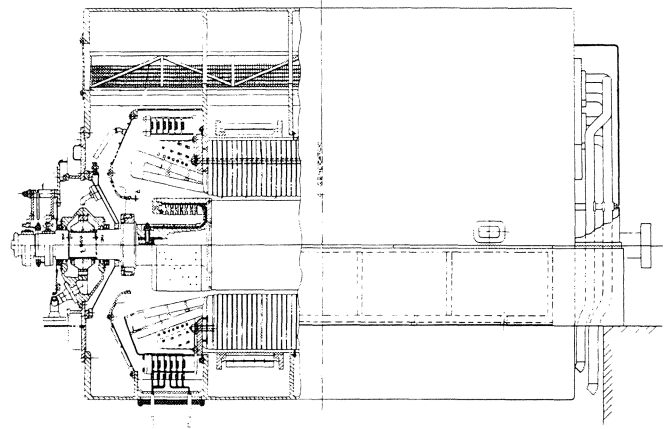


Figure 2. Generator Cross Section.

fluidized catalytic cracker (FCC) steam turbine came with a complete set of spare parts including spare diaphragms, nozzle block, and rotor. The value of the spares were exceptional for the company's purpose.

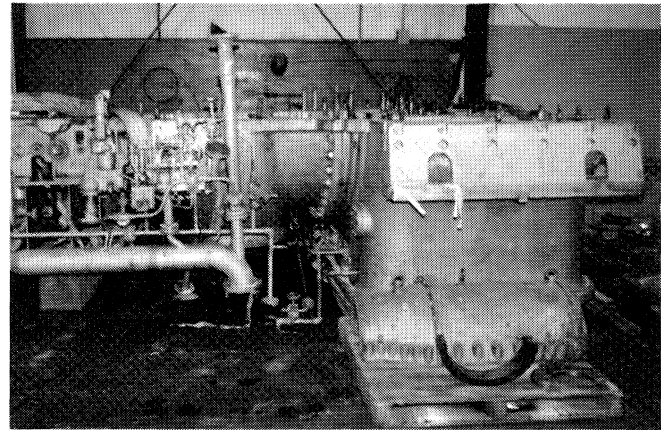


Figure 3. Turbine Lower Half Case as Purchased. Note existing governor relay system.

In December of 1987, the steam turbine case had been opened to allow internal examination. This occurred prior to purchase. The refinery was paid about \$5,000 to complete the work. This internal examination was required to develop a scope of repair for the equipment as well as for purchase approval.

The internal investigation revealed that the equipment was in excellent shape with the exception of the first five stages of the turbine. A leaking high pressure trip and throttle valve had admitted hot steam condensate into the front section of the turbine causing mild corrosion, rust, and pitting damage to the blading and rotor disks. The inspection was limited to splitting the turbine case and one bearing. No other equipment was disassembled at this time. The good condition of the rotor and case is essentially attributed to the fact that the refinery had kept up continuous warm oil lubrication and weekly shaft rotation from the date of shutdown until demolition and removal six years later. These practices minimized corrosion damage from the presence of the hot condensate.

Based on the foregoing inspection, a specification to repair the various components was prepared and sent to prequalified vendors. What follows is an overview of the repair/redesign of this T-G set with emphasis on some consideration in reapplication.

## ENGINEERING CONSIDERATIONS FOR REAPPLICATION

### Operating Constraints

Several operating constraints had to be included in the operating software for the turbine generator control. A number of off-design operating conditions potentially damaging to the steam turbine were possible. The first scenario was for an excess of steam flow to condensing. With a rated flow of 176,000 lb/hr a straight through to condensing, shaft power exceeds the generator's 15 MW rating. The governor, therefore, had to be programmed to monitor electrical output and limit steam flow, if required.

The possibility of induction of steam from the 50 psig header into the turbine condensing section had to be prevented. The 50 psig header often operates at temperatures only slightly above the condensation point. A large slug could destroy the turbine. To protect against this scenario, flowmeters would be required to ensure a positive extraction flow in excess of 20,000 lb/hr.

The turbine condensing section was also a cause of concern. Although the first extraction (sixth) stage would be modified for a maximum 170,000 lb/hr flow, the remaining stages were physically capable of passing nearly 350,000 lb/hr steam flow. Two potential problems existed. The first was that at low steam flow, below 50,000 lbs/hr (Figure 4), the blades would be moving faster than the steam. Moisture droplets formed in the last stages would cause rapid erosion and increasing drag losses. Removal of the last stage was considered, but rejected for the following reasons. At four inch HgA condensing pressure, the maximum flow that the remaining stage could pass without going into "shock" flow is only 130,000 lb/hr. At two inch HgA, which is the normal winter condensing pressure in this climate, the maximum flow would be only 65,000 lb/hr. Since the turbine was expected to operate as a spare for the existing 17 MW condensing turbine generator, 130,000 lb/hr flow was considered below the requirement for condensing capacity. In addition to flow restriction, removal of the last stage decreased overall efficiency significantly.

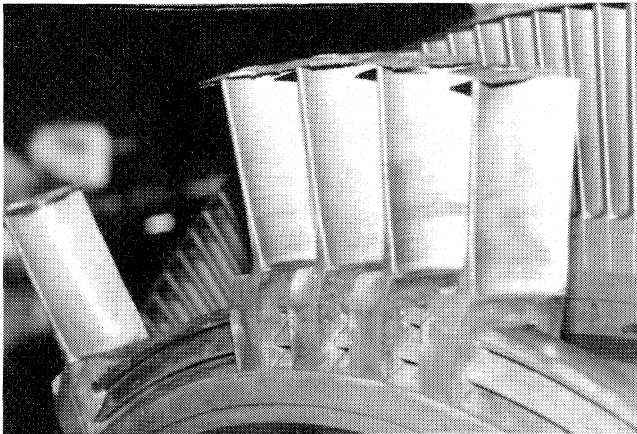


Figure 4. Sixth Stage Blading Before Modification in Removal Process.

The second problem has already been covered. With the last stage in place, the maximum exhaust flow without entering the "shock" regime (Figure 4) is rated at 260,000 lb/hr at 4 in HgA, (therefore, 130,000 lb/hr at 2 in HgA). Operation in the shock regime would cause unstable flow and subject blades to high fluctuating loads, possibly leading to catastrophic failure.

Control software for the turbine would have to be programmed to avoid the risky operating points already described. With an extraction flow below 20,000 lb/hr, the grid valve will automatically move to close to maintain a positive extraction flow. For exhaust end flow below 50,000 lb/hr, an alarm will be initiated. Note that a minimum flow (approximately 5,000 lb/hr) of cooling steam is always circulating through the exhaust section. To avoid entering the "shock" flow regime, an algorithm has been programmed to calculate the allowable exhaust flow based on the condenser pressure. The grid valve will close to keep from exceeding the calculated limit.

### Rotordynamic Analysis

Lateral and torsional analysis of the turbine generator were completed by an independent consultant. The primary purposes of the study were to identify resonance points within the operating speed range and to aid in coupling design requirements. The original plan was to have the rotors fully modeled prior to final coupling selection. The lack of available generator rotor dimensional data held up the analysis. Once the project funding was approved in July 1988, the generator was disassembled and rotor measurements were taken. The selected coupling's mass-elastic data were then entered to complete the analysis.

*Torsional Analysis* was done using rotor models developed for the lateral analysis. The TWIST computer program was used to calculate the torsional natural frequencies and mode shapes. A Campbell diagram was also generated.

The train operates between the first and second torsional natural frequencies. The first frequency is located at 858 cpm and the second was at 12,588 cpm. These frequencies are significantly separated from all low multiples of operating speed. Excitation forces at higher harmonics are considered insignificant for direct coupled turbine generators sets. No torsional problems are anticipated and the train fully meets API 612 torsional specifications.

*Steam Turbine Lateral Analysis* was done from a computer model built from measurements taken manually. The study was done using original design clearances for the sleeve bearings. Several computer programs were used to complete the analysis. Bearing programs used were for the calculation of stiffness, damping, power loss, operating position and rigid rotor stability for plain sleeve and pressure dam bearings. Rotor bearing models were used for calculation of undamped critical speeds, for forced response analysis and generation of damped natural frequencies, mode shapes and log dec values.

The undamped critical speed analysis indicated that support stiffness greatly influences the location of the criticals. As the bearings get stiffer nodal points move toward the bearings. This decreases damping, a function of the relative rotor motion. Critical speeds were calculated. The first critical had split peaks at 1950 cpm and 2550 cpm, due to the fact that the sleeve bearings were much stiffer in vertical than in the horizontal direction. Amplification factors for each peak were acceptable at 7.9 and 4.0, respectively.

The second critical was calculated to peak at 3600 cpm, right on the running speed. The amplification factor 2.3, indicated a well damped bearing mode. Even though it was well damped, the consultant recommended moving the second critical off of the operating speed by changing from sleeve to tilt-pad bearings. Tilt-pad bearings would change the critical speeds to 2300 cpm and 6000 cpm, outside of the API suggested separation margin.

This advice was not followed. Instead, sleeve bearing clearances were reduced from 0.022 in to 0.009 in on six inch diameter steam end and from 0.017 in to 0.012 in on the eight inch diameter exhaust end. This change was calculated to increase

bearing stiffness and, therefore, increase the second critical speed while moving the two first critical peaks closer together. As a precaution, tilt-pad bearings have been designed and can be built and installed on short notice, if required. Field fast Fourier transform (FFT) analysis should provide enough information to make a proper evaluation of the situation.

*Generator Lateral Analysis* was also done from a model developed from measurements taken on the generator rotor. Modelling the generator rotor without detailed knowledge of rotor construction or bearing support stiffness proved to be a problem. Critical speeds were split at both the first critical (1600 and 1925 rpm) and the second critical (3800, 4950, and 5650 rpm). The only concern was that the first peak is within the API separation margin. Fortunately, it is very well damped with a 2.7 amplification factor (AF).

The bearing stability analysis indicated that the first mode was just below the stability threshold at a  $-0.11$  logarithmic decrement. The consultant recommended a conversion to tilt-pad bearings. This would ensure stable operation as well as moving the 3800 rpm peak out of the API separation margin. After much internal discussion, this notion was rejected, based on the 20 years of troublefree operation experienced by the original owner. Vibration monitoring probes were added to the bearings to alarm and shutdown the generator in the event of an unstable whirl. Tilt-pad bearings were designed for manufacture, if ever required.

#### Turbine-Generator

As previously stated, a major obstacle was excessive blade stress in the stage after extraction, stage six. The ideal solution would have been a new redesigned stage, however, due to time constraints, modification was the chosen alternative. The extraction stage was modified by shortening the blades and welding specially machined inserts into the diaphragm to maintain efficiency while reducing blade stress. Performance and blading studies were conducted on these modifications (Figure 4).

The generator and steam turbine were not a matched set. A new dry coupling was selected and installed to replace the original geared coupling (Figure 5) on the steam turbine and the solid coupling originally used on the generator. There were also many minor changes such as the addition of vibration and temperature probes required to bring the entire package up to current API standards.

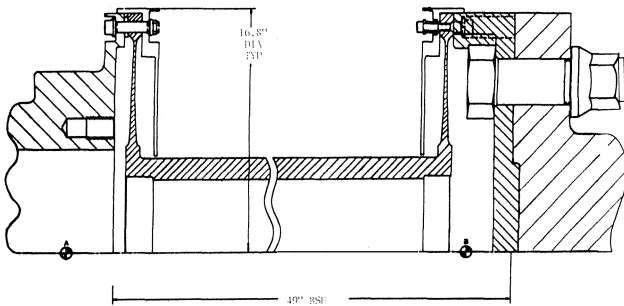


Figure 5. Coupling Design.

The project included demolition and equipment recovery from the center of an operating unit at the Louisiana refinery. In addition to the turbine, the huge surface condenser, trip, and throttle valves and other miscellaneous components were salvaged. Asbestos insulation, common on most older installations, provided additional complexity to an already difficult job. A new API-614 lube oil console had to be manufactured.

#### Redesign Turbine

Two approaches were considered during the initial engineering review, the manufacture of new hardware for the rotating and stationary components of the induction/extraction stage, versus the adaptation or modification of the existing hardware.

Competitive bids were solicited from selected "prequalified" vendors. Final selection took many factors into consideration. Of course, price and delivery, *after meeting* all engineering requirements, became the factor which decided the issue.

Deliveries of "all new" hardware would not fit into the project's timetable. Further pricing for new hardware was weighed against the "redesign of existing." The selection of new hardware implied new blading and disc at the extraction stage, plus a new mating stationary grid valve diaphragm. Destacking of the rotor would further impact the rerate price.

Original performance curves for the turbine were available.

The task, therefore, became to convert the subject turbine from induction to automatic extraction operation at changed steam conditions and produce expected performance curves and background information.

Detailed measurements of the turbine steam path were made, data gathered and reviewed to ascertain (and in some cases confirm) the basis of the original performance maps some of these were:

#### Original Equipment Data

- Technical manual
- Flow-hp induction chart
- Cross section assembly
- Model number
- Rating hp rpm
- Inlet nozzle size
- Induction nozzle size
- Exhaust size
- Steam conditions
- Outline
- Rotor assembly drawing
- Grid valve stage (Figure 6)
  - number of nozzles/number per 360 degrees
  - nozzle height at outlet
  - nozzle throat at base/tip
  - number of grid valve ports and number of nozzles under each
    - rotating blade profile and width
    - rotating blade throat at exit (base and tip)
    - rotating blade height (active)
    - rotating blade fastener description
- Stage following grid valve stage (same date as immediately above)
  - Diameter at base of rotating blades (both stages)
  - Grid valve details

These measurements were reviewed for confirmation of original design performance and for prediction of capability when operating with the new "conditions of service" (COS).

	<i>Original</i>	<i>Modified</i>
• Rating hp, kw	20,900	15,000 kw
• rpm (3600)	3300 to 3650	3,600
• Inlet steam press/temp	600°/700°F	650°/750°F
• Extraction press at flange	40 psig/IND	55 psig/EXT
• Exhaust pressure	4 in HgA	2 in HgA

• Max required lbs/hr		
• Inlet flow	176,000	176,000
• Extraction flow	0	170,000
• Exhaust flow	200,000	176,000
• Minimum required lbs/hr		
• Inlet flow	Cooling steam	50,000
• Extraction flow	N/A	0
• Exhaust flow	Cooling steam	Cooling steam
• Normal required lbs/hr		
• Inlet flow	Cooling steam	170,000
• Extraction flow	-0-	80,000
• Exhaust flow	200,000	90,000

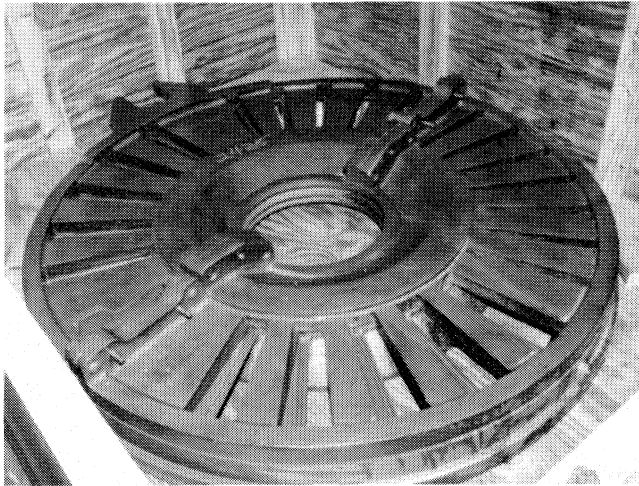


Figure 6. Grid Valve Assembly.

The redesign summary consisted of reviewing:

- Grid valve stage (stage six).
- Active nozzle height.
- Active blade height.
- Campbell diagram.
- Goodman diagram.
- Review of preceding and succeeding stages.

The parameters placed on the extraction stage (stage six) of critical impact were:

- Max flow to condenser lb/hr - 190,000
- Max extraction pressure, psig - 55
- Goodman Factor - hoped for <20

Shortening the blade height from 4.0 in to 2.13 in on the extraction stage suited all requirements but the Goodman factor—that being 15.7. A review of the “Goodman factors” for stage six assumed three points as representing the most severe operating conditions. These were:

- First valve section to open — the point where maximum energy drop occurs across the smallest arc of active nozzles.
- Maximum permissible flow at condenser exhaust pressure of two inches HgA.
- Flow equal to maximum predicted valves wide open (VWO) throttle flow at 190,000 lb/hr.

The desire to have Goodman factors exceeding 20 were offset by a need for a longer blade being able to pass more steam to the exhaust section during plant upsets. The generator and the electrical systems were both examined for system upsets, i.e., 190,000 lb/hr flow would produce 18 MW at the generator terminals.

The “Goodman factor” shortfall and steam flow needs could only be reconciled when viewed in context with the stage six “Campbell diagram.” Construction of the “Campbell diagram” revealed no interferences associated with those normally accepted as high for resonant stresses.

After review of all design goals, a reasonable compromise was reached and the rehabilitation of the turbine with new COS could go forward (Figure 7).

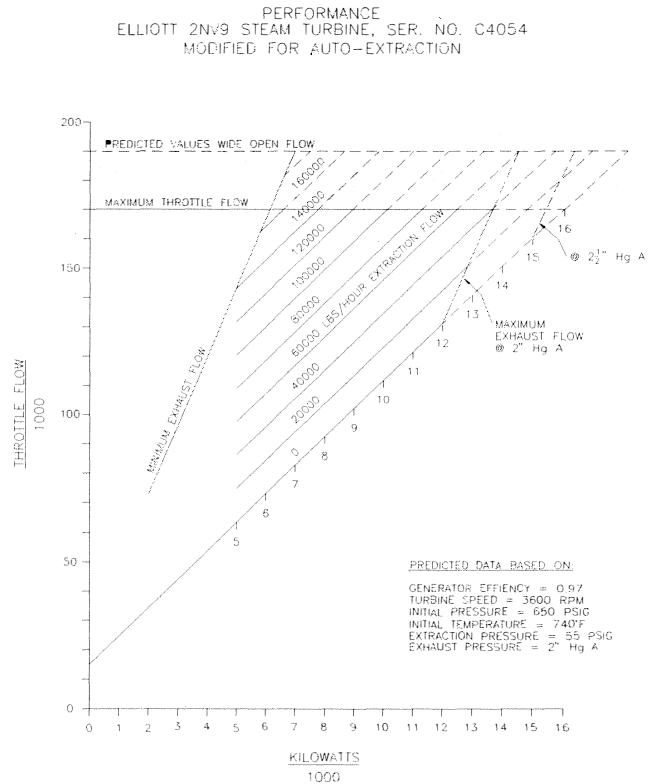


Figure 7. Performance Map After Modification.

Turbine Overhaul

The turbine overhaul proceeded with the usual repair activities. Original data was scarce or generic only, but this type of equipment is easily rehabilitated.

Nondestructive testing verified that the equipment was structurally sound and, except for normal wear and tear, the turbine was of “good stock.”

The turbine redesign study required that the stage six row of rotating blading and the mating diaphragm openings be shortened (reduction in area). Stage six rotating blades were marked for location and removed. Blade height reduction, incorporating new tenons, was machined by standard methods, including surface treatment with shot peening (Figure 8). Stage six blades were reinstalled, shrouds fit and tenons hydraulically upset. Ultrasonic testing and shroud trimming completed the stage six rotating blade revamp.

Modification to the stage six diaphragm was somewhat more difficult. Area reduction required installing three-dimensional fillers between vanes of the diaphragms (Figure 9).

Epoxy molds were made and tried in each vane opening. A compromise shape was finalized and a complete set was machined on a special purpose milling machine. Individual dressing on each filler piece allowed for very close fits; the fillers were secured by welding. Thorough nondestructive testing

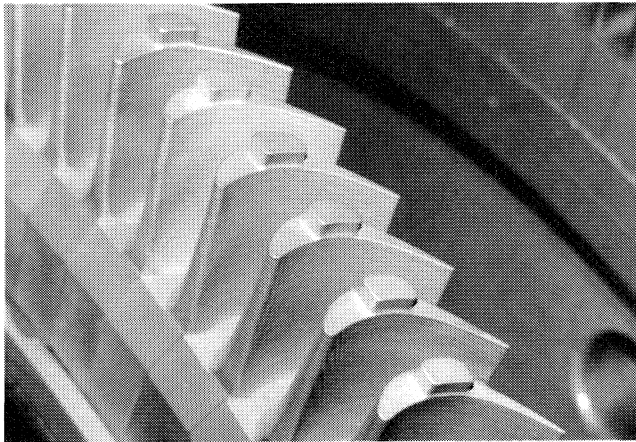


Figure 8. Sixth Stage Blading After Modification in Installation Process.

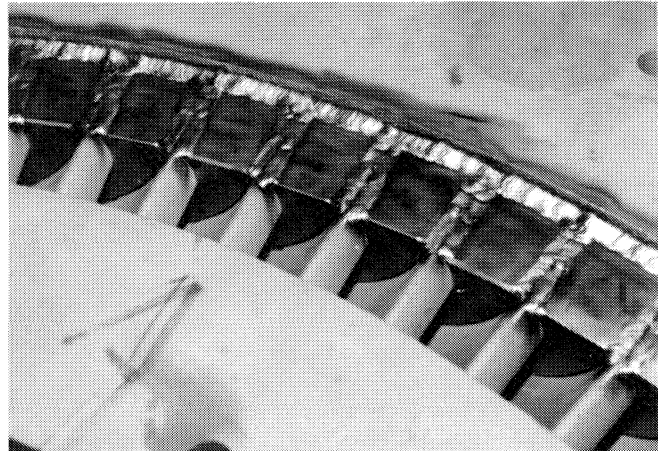


Figure 9. Sixth Stage Diaphragm After Filler Blocks Installed.

completed the grid valve diaphragm modification. Modifications to bearings/housings were made to allow for temperature and vibration sensing elements. Valve chest parts were renewed where required.

Contract and spare turbine rotors were balanced to API-612 requirements, including 12 point residual checks. Probe target areas were burnished and calibrated for electrical runout.

#### Coupling

The coupling selection received a great deal of attention. In addition to the previously mentioned lateral and torsional analysis, a coupling consultant was contracted to audit the design. The successful coupling bid was scrutinized for low windage, ease of maintenance, design reliability and, of course, cost and delivery.

This detailed analysis resulted in several suggested modifications to the manufacturer's standard design. Some changes, which were believed to be improvements, were reluctantly accepted by the vendor while many modifications were strongly resisted. The final contract design was a 49 in diameter between shaft ends (DBSE) diaphragm coupling with a custom adapter to match the existing generator's integral hub (Figure 5). The long coupling was required so as to provide adequate access room between the turbine and the generator.

#### Trip and Throttle Valves

The original trip and throttle valves were salvaged with the steam turbine. The high pressure trip and throttle valve is an eight inches Class 600 latch type while the low pressure, non-return valve (induction now extraction) is a 20 in Class 150 oil operated valve. The original intention was to overhaul the valves and reuse them. The valve manufacturer evaluated these valves for the new application as requested.

Concerning the eight inches high pressure inlet trip and throttle valve, the manufacturer stated that the salvaged valve was an old design and was no longer furnished for the plant's operating conditions. Although the valve was correctly sized for the new steam conditions, and was probably acceptable for FCC air blower service, they did not recommend it for use with a steam turbine generator set. Turbine generators commonly are expected to experience complete loss of load. The resultant rapid rotor system acceleration to overspeed trip conditions requires fast, reliable valve closure. An FCC blower turbine would see instantaneous acceleration only in the unlikely event of a severed coupling or fractured shaft. The salvaged valve's construction and materials were not as reliable for outdoor turbine generator service as with newer designs. The valve was the

push-to-close type, which have been somewhat notorious for not closing due to steam deposits, corrosion, etc.

A new valve was ordered, fully oil operated with exercisor, totally enclosed, with pull-to-close design. This new valve is suitable for outdoor installation and has many design improvements over the old valve. Unfortunately, the promised delivery was almost one year; several months after the planned startup. After much discussion, the existing valve was determined to be reliable for short term operation and was refurbished, totally tested, and installed. For safety, special care is being taken to prevent any external corrosion and manual exercising is done weekly to prevent internal deposit buildup.

The nonreturn, oil operated, valve was not only an old design, but a one-of-a-kind design as well. The manufacturer had no suitable equivalent replacement valve, although a quality hydraulic check valve would probably work in the service. The valve was originally designed for induction service; but, it will work in extraction service equally well. In the event of a turbine trip, the check valve must prevent 50 psig steam in the extraction header from being admitted into the turbine low pressure end. This valve was refurbished at an OEM-approved shop in Houston. New parts had to be made locally to support the overhaul.

In addition to the changed steam conditions, the trip valve operating oil pressures were also changed. The original lube and control oil system had three different pressure headers: 12-15 psig for bearings, 125 psig for servomotors and 85 psig for control oil to the governors and trip valves. Both trip valves were built for an operating oil pressure of 60 psig. Research confirmed that the valves could handle 125 psig oil pressure. The 85 psig header was deleted from the oil system design. The oil trip setting was kept at the same 60 psig decreasing and had no measurable effect on the trip time.

#### Governor Retrofit

The original governor used two old style hydraulic "jewel box" designs. According to its refinery operators, "it never worked well." It was scrapped in favor of a new digital governor system. "Off the shelf" digital governors were considered first. They were programmable, inexpensive and capable of controlling the extraction requirements. Although these modern governors are more than adequate for most applications, they did not meet the plant requirements for steam header control. In the plant's control schemes, the high pressure steam turbine inlet valve rack is the primary pressure controller for the plant 650 psig header. Upsets in 650 psig system pressure cause severe operating problems throughout the entire plant.



It was decided that all header pressure control equipment, the existing 17 MW turbine generator, the new 15 MW turbine generator, the 50 lb letdown valve, and the atmospheric vent header would all be strictly controlled and coordinated by the plant DCS computer (distributed control system). The logic effectively eliminates any "fighting" between the principal instruments in various states of calibration. In tandem with the inlet valve, it was also desirable that the extraction grid valve (influencing the 50 psig header) also be under DCS control.

Neither of the "off the shelf" digital governors was capable of acting as a slave to the DCS during normal operation as dictated by the plant control philosophy. The simpler "off-the-shelf" governor systems had to be abandoned for a custom designed unit.

The new criteria specified a governor that would pass through 4-20 MA signals from the DCS to electro hydraulic actuators powered by the 125 psig control oil header. The "slave" status to the plant DCS would be active while the generator was in the normal operating mode exporting electricity to the utility grid. The governor automatically transfers complete control to the DCS once both tie and load breakers have closed and drops into a monitoring mode.

In the event that the generator should suddenly become isolated from the utility grid, the governor would sense the break and take control. The governor is set to control speed and, therefore, load until re-synchronized to the grid. The generator would be run in parallel with an existing 17 MW steam turbine generator controlled by a Woodward 2301 operating in the Droop mode. In the event that the generator breaker trips open the governor is required to take control and maintain speed at plus or minus five percent of 3600 rpm with only three or less speed swings. Once the governor takes control, signals from the plant computer are ignored.

The new governor is a Woodward 501. It has two redundant, proximity probe, speed pick ups. Two 4-20 milliamp current inputs for inlet and extraction valve position are fed from the plant computer into the governor and then to the electrohydraulic actuators which control the hydraulic servo motor valve positioners. The loss of either of these signals will initiate a trip condition. Generator load is monitored and is the third input in control scheme. Contact position signals were also required for the tie breaker to the utility bus as well as a load breaker for the new generator.

The 501 has all of the other normal function inputs, such as start, stop, reset, raise speed, lower speed, and three speed set-points for use in startup and slow roll. Electronic overspeed trip set points have been incorporated into the governor control. The redundant electronic trips are in addition to the mechanical trip.

## REWORK GENERATOR AND EXCITER PROBLEMS AND SOLUTIONS

A large problem was the severe lack of documentation, drawings, and vendor literature. The immediate need was for rotor dimensional data so that a lateral/torsional analysis of the turbine generator prior could be completed to ordering a new dry coupling.

Requests for drawings from the OEM were made in February 1988. Original drawings that were available with the used equipment were generic. Only one generator "general arrangement" drawing had dimensions. The OEM was able to produce another rotor drawing. However, it was wrong. A second drawing, more accurate, but incomplete, was produced later. The manufacturer stated that they were in the process of locating the file on the generator in their vault and it would only be a matter of weeks until the drawings were in the company's hands. Weeks

passed into months until the OEM finally stated all records for the generator had been lost or destroyed.

### Generator Overhaul

The generator is a synchronous AC unit rated at 15,000 kW at 0.85 power factor, 14.4 kV, and 3600 rpm. The generator uses axial fans at each end to circulate air through the rotor and stator via two horizontally mounted water cooled air coolers and thus back to the fans suction. The generator is a 1968 vintage and was originally driven through a solid coupling by a gas turbine.

The initial visual inspection was very encouraging. The generator exterior showed only surface rust. An end bell was removed and the bearing cap lifted to reveal a bearing in the "as new" condition. Removal of the end bell offered limited access to the internals. No signs of corrosion or other damage were seen, in spite of the unsheltered outdoor storage. The lack of any rust or water in the bearing was very encouraging. The shop overhaul was expected to be straight forward. It was at this time the plant engineers discovered the rotation of the generator was clockwise, and had to be changed to match the counter-clockwise rotation of the turbine. This proved to be a simple matter due to the symmetry of the stator and rotor. Fan hubs and bearings were swapped from one end to the other to accommodate rotation reversal.

A scope of repair was initiated for the generator based on the initial inspection. The general scope was to simply disassemble, test, clean, reassemble and test again. This scope was sent out for bids to "prequalified" generator repair shops.

The workscope included the following items for inspection, test, and repair.

#### Inspection

- Visual inspections
- Dimensional inspection and documentation
- Cooler-hydrotest
- Rotor balance with 12 point residual unbalance check
- NDT-rotor retaining rings and all suspect parts
- Electromechanical runout checks

#### Testing (per IEEE Standard 56-1977)

- Partial discharge test
- Winding resistance of all items
- Insulation resistance of all items
- High potential test of all windings to ground
- All windings for shorted turns per IEEE Standard 522-1977

#### Repair

- Clean all parts
- Replace all bolts, gaskets, shims, etc.
- Modify bearings and fans for CCW rotation
- Spray and brace windings
- Add X-Y vibration probes to the bearing housing
- Fabricate new baseplates
- Preparation for shipment and outdoor storage

The actual scope of repair contained considerable detail. Any repair outside of the framework of the scope was to be referred to B.P. Chemical for consultation and authorization to repair. The exciter was outside of this scope of repair and was quoted separately.

Once the generator was disassembled, two minor problems surfaced. Leads from the stator resistance temperature detectors (RTDs) were corroded and had to be replaced. On the field, some minor checkout and refurbishment was necessary. Once the retaining rings were removed, minor corrosion to three rows of wedges and degradation of winding insulation on the leads

were visually apparent. Suspect wedges were removed and tested. They passed NDT and were cleaned up and reinstalled. Field winding leads were tested and insulation wrapping degradation was confirmed. These leads were rewrapped and retested. All of this work was in addition to the original scope of repair.

The plant operators utilized an inspection consultant, the same person contracted to oversee the turbine assembly inspection, and for generator "hold point" inspections whenever the plant electrical and mechanical engineers were unavailable. Having an inspector locally available helped to streamline the inspection process, as well as reduce the demand on plant engineering. For instance, the generator field failed balance witness checks on three different days for various reasons. Time that would have been lost travelling to and from Houston was saved while overall quality assurance was maintained.

#### *Exciter Overhaul*

The original exciter was a static system with slip rings and carbon brushes. All that remained of the exciter were the rings, brush holders and support assembly. The scope of supply for the excitation system asked for a retrofit with a brushless system. The plant operators requested a solid state power supply, field ground detection, loss-of-potential protection, over-excitation protection, and an external control interface system for adjustment of the target power factor. The unit was to be suitable for outdoor mounting in a Class 1, Group D, Division 2 environment.

The generator OEM flatly stated that they did not have the ability to retrofit this unit with the excitation system. Another OEM had the ability, but, declined the work. Finally, a third OEM quoted a brushless system; but, after they were awarded the repair contract, they recanted stating they could not put a brushless exciter on the generator. We resigned ourselves to having to face the continuing nuisance of maintaining the brushes. Still later, the successful bidder subcontracted the static exciter supply to another company that we had purposefully not solicited during the bid stage.

It should be noted for those who may be considering a used generator revamp that the OEM's engineering groups handling exciter design are virtually a different company from that same OEM's repair center. Lack of cooperation between the repair center and their engineering department made the job more difficult and caused numerous delays with general confusion. It was like dealing with two different vendors.

The generator revamp included a new housing for the brush holders. A custom designed purge and vacuum system was fabricated and installed on the exciter housing at site. This system will hopefully eliminate problems with carbon dust buildup.

#### CONCLUSION

Used equipment is one avenue available to the engineer faced with "impossible" schedule requirements or limited capital resources. Used equipment can often be upgraded and modified to meet current industry standards at relatively low cost. The used parts market, along with various after-market repair shops, are able to replace damaged parts or supply spares in the event

that the OEM no longer makes the part or cannot meet the required delivery.

Disadvantages of used equipment are also plentiful. In many cases, the equipment has not been "mothballed" correctly. In addition to corrosion, the equipment may not have been in good condition when it was shut down. Used equipment should also be inspected and repair costs estimated to avoid spending a fortune trying to rehabilitate equipment that should stay in the junk pile. Lastly, the availability of vendor information, performance curves, clearances, etc., should be considered.

Overall, most of the foregoing problems can be overcome. In addition to the OEM's, there are several qualified repair shops and engineering consultants capable of offering experienced assistance. On this particular project, the total cost for equipment, overhaul, engineering, construction and start-up was under \$3.3 million with startup only 12 months after purchase. Original estimates for the all new equipment option came in at close to \$10 million with a 18-24 month completion schedule.

#### *Startup and Operation*

The startup went very well mechanically. Electrically, the exciter has not produced the required kilovars (kvars). The plant to import VARS from the utility until the exciter can be fixed. The turbine generator startup was done, in fact, on an emergency basis, because the existing 17 MW generator tripped offline due to a bearing failure.

The 15 MW turbine generator is currently running on a straight condensing mode at 15 MW with zero extraction, while the 17 MW unit is being repaired. The 15 MW unit runs well with performance in the 10.5lb/kW range.

The only problems have been in the area of vibration. None of the resonance problems predicted by the lateral analysis have appeared; however, two other vibration problems have appeared. The first is on the turbine exhaust end bearing. Based on a high  $2\times$  amplitude on an FFT spectrum and a flat elliptical orbit, poor alignment is the suspected cause. A hot alignment check and realignment, based on accurate instead of assumed thermal growth, should solve this problem.

The second problem is exciter end vibration. Vibration on the exciter end bearing was very high, increasing with load and/or kvars. The overall amplitude reached 4.0 mils at full load. This was considered high since most of the plant equipment operates at 1.0 mil or less. The generator manufacturer, however, was not concerned and suggested that 3.0 mils was considered good and alarm should be set at 5.0 mils.

All of the excitation appears at  $1\times$  running speed and increases in amplitude whenever load (kW) or excitation (kvars) are increased. This problem is considered to be a thermally induced bow in the exciter area of the shaft producing an imbalance. Load and excitation increases raise the temperature of the shaft under the slip rings. Once the 17MW unit is back online, the plant plans to decrease the vibration amplitude by reducing the thermal effect with balance weights.

After correcting for the vibration problems, the unit will be brought up and operated with extraction, as originally planned.