OPERATING EXPERIENCE WITH IMPROVED STEAM TURBINE PACKING RINGS

by

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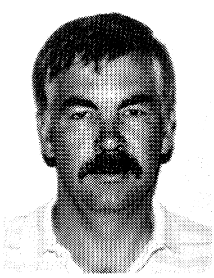

Steve Williams graduated from George Washington University in 1977, with a B.S. degree in Mechanical Engineering. He is a Registered Professional Engineer in the State of Maryland and a member of ASME.

He has been employed with Potomac Electric Power Company for 16 years. For the past seven years, Mr. Williams has been in charge of the Central Diagnostic Team which is responsible for performance testing and diagnostics of steam turbines and auxiliary equipment. He also spent several years working at Chalk Point and Benning Generating Stations as a Plant Performance Engineer.

Mr. Williams has been actively involved in the EPRI project to evaluate Power Plant Performance Instrumentation Systems at PEPCo's Morgantown Generating Station.

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Since joining PTI, he has participated in a wide variety of thermal plant optimization projects. He has also refined his long standing specially regarding performance problems of steam turbines. This work entails diagnosis of internal steam path problems from test results; careful inspection of disassembled turbines; analyses identifying root causes of poor efficiency; economically sound recommendations for repair; and suggestions for improved operating practices to prevent recurrence of the deduced losses.

Mr. Brandon is an instructor in PTI courses dealing with power plant performance and steam turbine optimization techniques.

In the course of his professional work, Mr. Brandon has been awarded eight patents. He is a Registered Professional Engineer in the State of New York, and a Senior Member of ASME. He has a number of professional publications.

ABSTRACT

The losses in turbine performance due to excessive leakages in the labyrinth seals can result in significant heat rate degradation to the turbine cycle. The modified packing arrangement described helps eliminate the rubs that occur during startups, which is the major cause of excessive clearance.

This packing was first installed at two utility generating stations. The performance of these units were monitored to evaluate the packing and these results are discussed in detail.

INTRODUCTION

Potomac Electric Power Company (PEPCO) has historically had problems maintaining proper clearances in their steam turbine seals. This has resulted in significant degradation to turbine efficiency and unit performance. The packing modification installed on the Dickerson Unit and later on the Chalk Point Unit has resulted in better control of steam seal leakage. This has resulted in significant improvement in unit performance which are described in detail herein.

TURBINE PACKING

A turbine is designed to convert the potential energy in steam into kinetic energy to produce useful work. In order to obtain this kinetic energy, high differential pressures are present throughout the turbine to accelerate the steam through the numerous turbine nozzles. The steam will attempt to take any path it can find into the lower pressure chambers of the turbine. Unfortunately, leakage paths around the turbine nozzles result in a loss of kinetic energy, and often result in an even greater loss where it disturbs the flow path upon reentry into the lower pressure section. The manufacturer attempts to control the leakage very closely, in order to maintain good turbine performance.

The basic method used by the turbine manufacturer in sealing the rotating shaft from the stationary parts of a turbine uses spring-backed labyrinth packings. These packings restrict the amount of steam leakage flow for a given pressure drop. The teeth of these packings are arranged in a high/low manner similar to that shown in Figure 1. This design causes the steam passing through the packing to encounter the back of the next tooth resulting in eddies, which destroys the kinetic energy of the steam. The steam must then accelerate pass the next tooth and this process continues through all the packing teeth which considerably restricts the amount of flow passed for a given clearance.

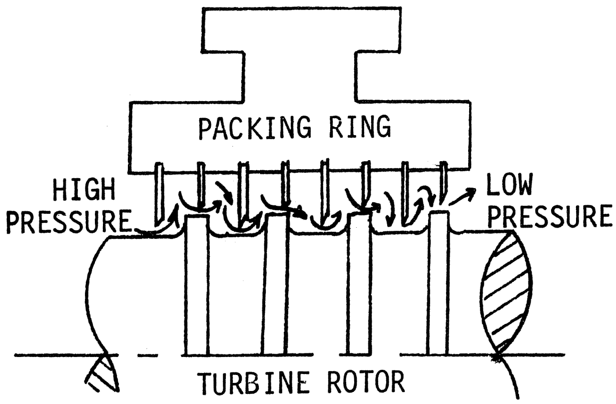

Figure 1. Turbine Packing—Labyrinth Seal.

The packing ring holds the teeth in proper position relative to the shaft and usually has four or more teeth per ring. The clearance of the packing is determined upon assembly, by appropriate machining of the packing ring shoulder or the tooth length. This clearance is maintained by the spring clip which holds the packing ring shoulder against the ring holder. Design clearance is approximately 0.001 in per inch of shaft diameter. This is normally 0.030 in for a typical turbine design. It is also important to maintain the sharp edge on the tooth to minimize the leakage effect.

The packing ring (Figure 2) consist of several segments around the circumference which allows the packings to move independently, while the spring clips maintain a tight shoulder fit.

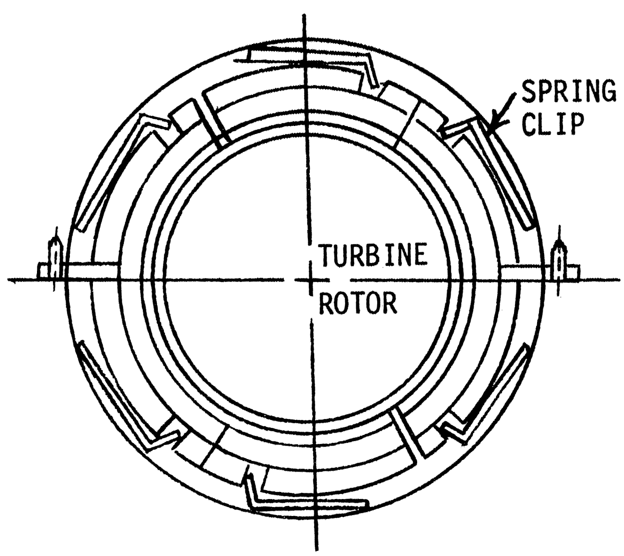

Figure 2. Packing Ring.

The flow of steam that passes the packing is calculated by Martin's formula Equation (1).

$$W = K \times A \times B \times (P_1/V_1)^{1/2} \qquad (1)$$

where

W = flow
K = flow constant
A = effective area
P_1 = upstream pressure
V_1 = upstream specific volume
B = a function of the number of teeth and the downstream to upstream pressure ratio

$$\text{Effective area} = \text{clearance} \times \text{diameter} \times \text{pi} \qquad (2)$$

After the turbine is designed and built it is normally recognized that the only remaining controllable parameter is the shaft clearance. Holding all parameters constant, except clearance and combining Equations (1) and (2) reduces Martin's formula to:

$$\text{Leakage flow} = \text{constant} \times \text{clearance} \qquad (3)$$

Using Equation (3), it becomes clear that if the clearances are doubled then so is the leakage flow. The problem is made worse if the sharp edge packing tooth is blunted, resulting in even larger flows due to increasing the flow coefficient (normally reflected by a change in k).

PROBLEM

The ability to maintain tight clearances at Dickerson has always been a problem as it is in most other turbines. During startup, the turbine is susceptible to vibration as the rotor is brought through its critical speeds. The end bearing vibration may not be excessive, but the center span is subject to large defection due to the long lengths of unsupported shaft. Dickerson units have no shaft prewarming capabilities, which adds to the shaft bowing problem.

The inner shells, diaphragm and packing boxes which hold the stationary packing rings are subjected to large temperature differentials during startup. This condition will result in the normally round packing ring taking on an egg shape. This condition can lead to distorted clearances which may result in a packing rub.

The manufacturer designs the packing segments and spring clips to move if a shaft rub occurs. This design is intended to save the turbine shaft from excessive cutting, but will not prevent damage to the relatively soft packing teeth. The resulting condition of shaft rubbing consequentially produces heat, possibly bowing the shaft, increasing the intensity of the rubbing, causing increasing clearance and also causing the packing teeth to become somewhat embrittled which makes them susceptible to breakage and even larger clearances.

The bowed rotor condition rubs the labyrinth packings and also the tip seals designed to prevent leakage around the rotating buckets. This tip seal leakage loss often exceeds the losses found in the labyrinth packing.

If packing clearances could be opened at startup and closed during normal load operation most severe packing rubs could be avoided.

MODIFIED DESIGN

The modification to the Dickerson Unit was designed to accomplish the objective of increased clearance at startup and tight clearance at full load. The original packing design was field modified to accomplish this objective during the Fall 1986 overhaul.

The original rings of packing are shown in Figure 2 with the spring clips inserted. The major change required is to remove the spring clips, which force the packing ring toward the shaft and install coil springs which force the packing rings away from the shaft (Figure 3). The springs size and depth of the holes drilled in the packing rings are predetermined based on the expected pressures around the packing.

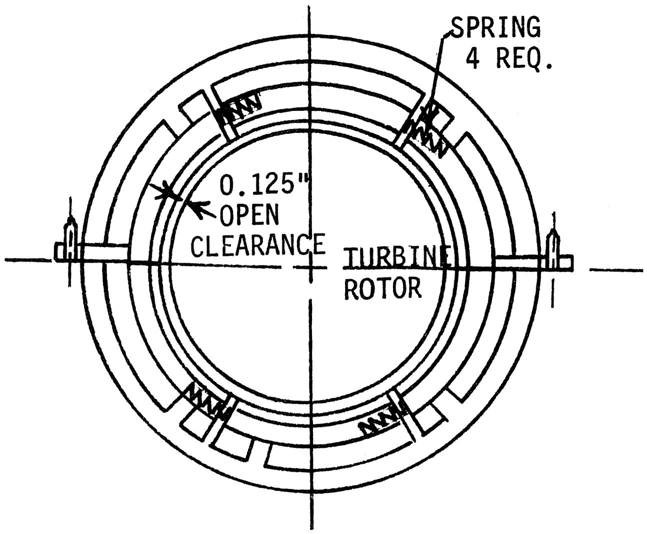

Figure 3. Modified Packing.

The basic theory used in modifying the packing is that the pressure forces behind the packing increases with the units' flowrate and can be used to overcome the spring and friction forces acting on the individual packing segments. By proper design of the coil springs, the pressure forces acting on the packing can be utilized to cause the packing to move from a large clearance to a small clearance at a predetermined flow condition that is known to be beyond the condition where severe rubbing is likely to occur. This method then determines the unit load at which the packing rings close and the tight clearances are established.

The static pressure forces acting on the cross section of packing are shown in Figure 4. By using a vector diagram similar to that in Figure 5, the forces can be calculated for a given load and the springs sized to balance the upward and downward forces. This oversimplifies the problem, since the friction forces act against the closing force. The determination of the friction coefficients was part of the preliminary research work partially sponsored by the Department of Energy. Also, the pressure drop across the packing teeth from P_1 to P_2 is somewhat non-linear and further complicates the analysis.

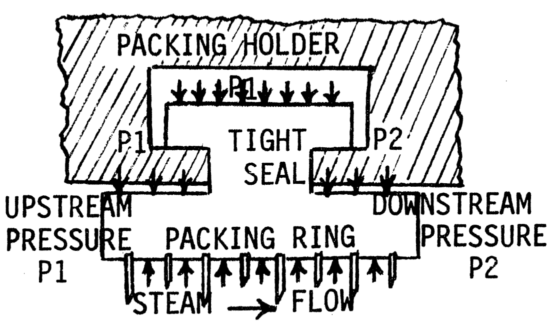

Figure 4. Pressure Forces.

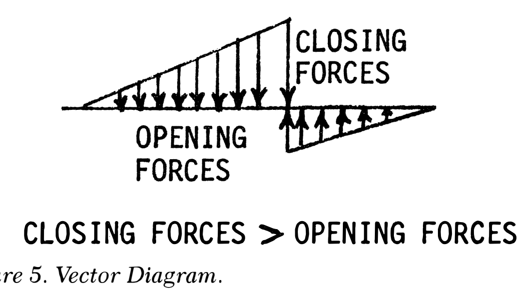

Figure 5. Vector Diagram.

The packings are held open until the predetermined unit flow (five to twenty percent) is obtained. The open clearance of approximately 0.150 in is determined by the back of the tooth ring fit with the housing. This is sufficient to avoid normal startup problems and at the predetermined flow the packings close to normal running clearances of 0.025 in.

The pressure forces to close the packings were felt to be marginal on numerous rings. To resolve this problem, the design required removal of several of the last short teeth, and this resulted in less force holding the packing open (Figure 6). Fewer packing teeth resulted in increased leakage flow, but are far offset by the improved clearances being maintained.

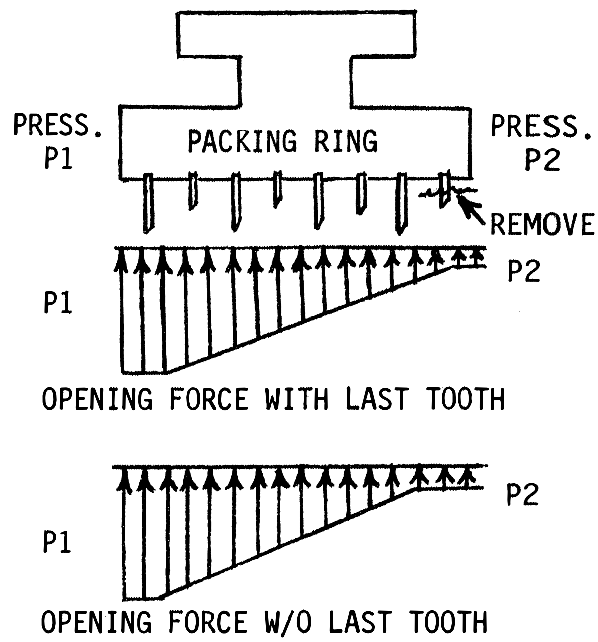

Figure 6. Effect of Tooth Removal.

PRELIMINARY TEST

The new packing design was developed and patented by Ronald Brandon, of Power Technologies, Incorporated. The preliminary work was sponsored by the Department of Energy and New York State Energy Research, which included test work on a prototype packing assembly, followed by installation in a boiler feed pump turbine at Niagara Mohawk.

Preliminary test work included cycling studies for reliability and development of the friction coefficients. After the preliminary investigation was completed the decision was made by the utility company to install the packing in the Dickerson Unit during the Fall 1986 overhaul.

PERFORMANCE FACTORS

A variety of benefits were expected to result from the effects of the improved packing. These included:

- *Decreased stage packing leakage*. This creates a direct improvement on the high pressure (HP) or intermediate pressure (IP) turbine section efficiency.
- *Decreased tip (or shroud) leakage*. This benefit results from the avoidance of bowed rotors (normally caused by packing rubs)

that result in damaged tip seals. This saving can be expected to even exceed the benefits of the improved diaphragm packing clearance.

• *Decreased (number two packing) N-2 leakage flow.* Flow leaking through this critical six rings of packing which separates the first high pressure stage from the first reheat stage (Figure 7), will bypass all but the first HP stages, causing direct kilowatt (kW) losses. In addition, the leakage flow will reduce the IP section bowl enthalpy with a resulting reduction of reheat (RH) section available energy. This latter loss can often be about 50 percent as big as the HP loss.

• *Increased first stage shell pressure.* It is common to find the turbine first stage shell pressure lower than design by from five to fifteen percent. This has the effect of taking energy off the relatively efficient later HP stages and putting more energy on the first stage (the least efficient stage). The added energy drops the already low first stage efficiency even lower. It was expected that the improved packing would improve first stage shell pressure to be closer to design values.

• *Decreased excess flow capacity.* Excessive leakage has a secondary effect of increasing the turbine flow capacity. While this may have some side benefits, it also causes some bad HP efficiency effects across the load range by requiring increased throttling of control valves for any normal level of flow. This causes significantly poorer HP section efficiency. Note that where the excess flow capacity is of value, it may still be available by way of increased initial pressure—five percent overpressure being commonly acceptable on most turbines.

INSTALLATION

The modified packing was installed in the N-2 packing and in all the HP-IP diaphragm packings (Figure 7). This included seven diaphragm rings in the HP, five diaphragm rings in the IP, and six N-2 rings. The rings were being replaced as a part of the Dickerson Unit 3 overhaul; therefore, the cost did not include the rings which were standard packing. Additional cost included the purchase of the springs, machining the spring holes, lapping the seal joint and cutting slots to assure the back of the packing was pressurized. The field work involved cutting the teeth and shoulders to obtain the desired closed clearances. This is a normal company overhaul procedure used with stationary packing.

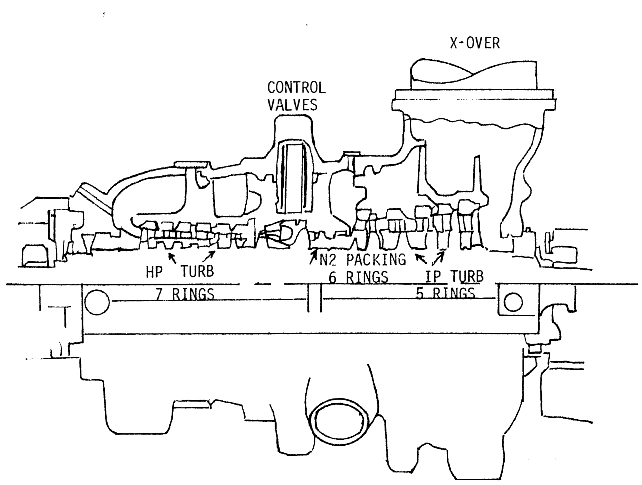

Figure 7. Turbine Cross Section.

ONLINE ANALYSIS

The utility company decided to conduct online performance monitoring to evaluate the new packing design. Steam pressures and temperatures were being monitored around the HP and IP turbines and the data was stored on a desktop computer.

The unit was continuously monitored through the end of 1987, and includes several startups which exercised the packing opened and closed. The unit was also tested under controlled conditions using precision instrumentation to verify the results obtained with the on-line monitor.

PERFORMANCE RESULTS MONTHLY FUEL HEAT RATES

The 12 month running unit heat rate averages for the three Dickerson units is shown in Figure 8. Unit 3, with the improved packing, shows a consistent improvement, which is expected to level off at about 200 Btu/kWh. Most of this difference is credited to the improved packing, but other improvements made during the inspection period also contributed to the improvement.

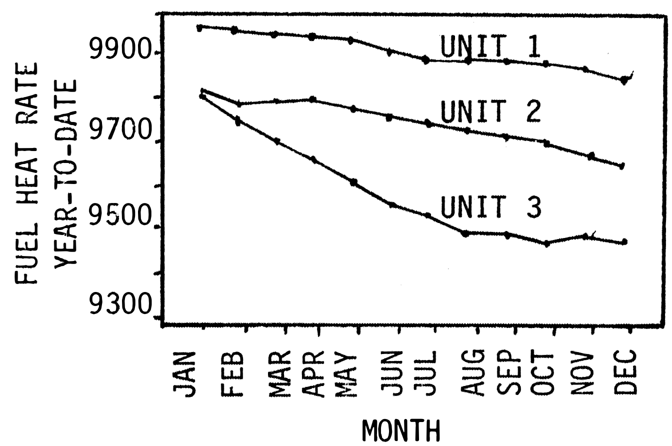

Figure 8. 1987 Monthly Fuel Heat Rate.

HP SHAFT/LP SHAFT MEGAWATTS

The Dickerson units are cross compound. Therefore, a comparison of HP shaft to low pressure (LP) shaft output ratio can be made after correcting for standard conditions. The following results have been obtained.

Date	Megawatt Ratio
Design	1.045
July 85 (pre-overhaul)	0.97
December 86 (post-overhaul)	1.046
June 87	1.040
November 87	1.055

This ratio is a good indicator that the HP-IP performance is being maintained. It is significantly better than the pre-overhaul ratio of 0.97. For a given throttle flow rate, the HP/IP KW output is up by about 5000 kW.

HIGH PRESSURE TURBINE EFFICIENCY

A plot of HP turbine efficiency, (taken at number four control valve crack point) over the twelve months test period since the overhaul is presented in Figure 9. The 81.5 percent efficiency calculated shortly after startup is 1.5 percentage points higher than the best efficiency ever obtained on any Dickerson Unit in the last seven years.

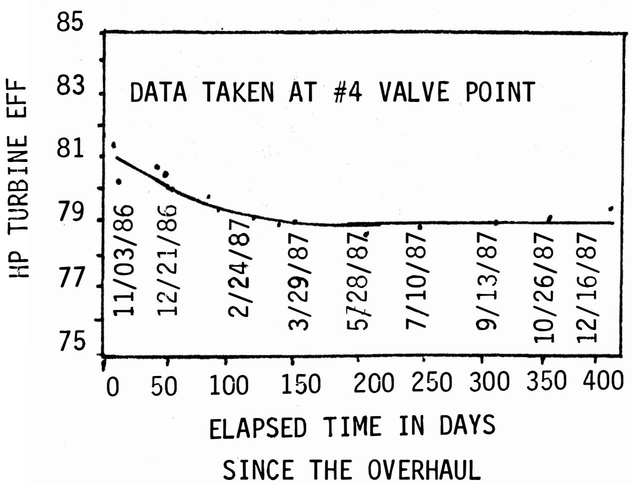

Figure 9. High Pressure Turbine Efficiency Trend.

Unit 3 HP turbine has degraded 2¼ percentage points since the overhaul. The most likely cause of the degradation is due to weld bead damage, deposits and possibly some erosion. Based on previous overhaul inspections, not all of the degradation can be assumed to be due to the above mentioned items, so some degradation must be due to packing and excess seal leakage. How much is indeterminable. It should be noted that the Brandon design is for packings only, therefore, spill strips are still vulnerable to shaft rubbing, although to a lesser degree. A plot of the June 1987 enthalpy drop test and a turbine test run in 1985 is depicted in Figure 10. As stated earlier, the turbine efficiency has dropped since the startup in October, but it is still above the 1985 turbine performance level. The design efficiency curve is a valve best point type curve and, therefore, it does not show the valve loops as the test data shows.

INTERMEDIATE PRESSURE TURBINE EFFICIENCY

Trend data was found to be misleading in evaluating the IP turbine performance. This is a result of poor instrumentation

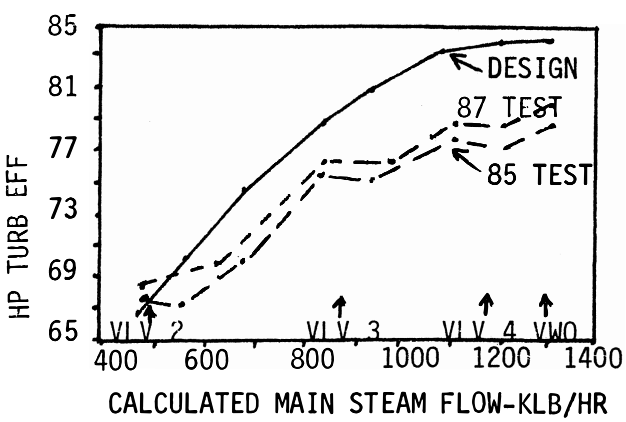

Figure 10. HP Efficiency Test.

and non-steady state conditions. As a result, a controlled test was conducted using calibrated resistance temperature detectors (RTDs). Apparent IP turbine efficiencies were calculated and are shown in Figure 11.

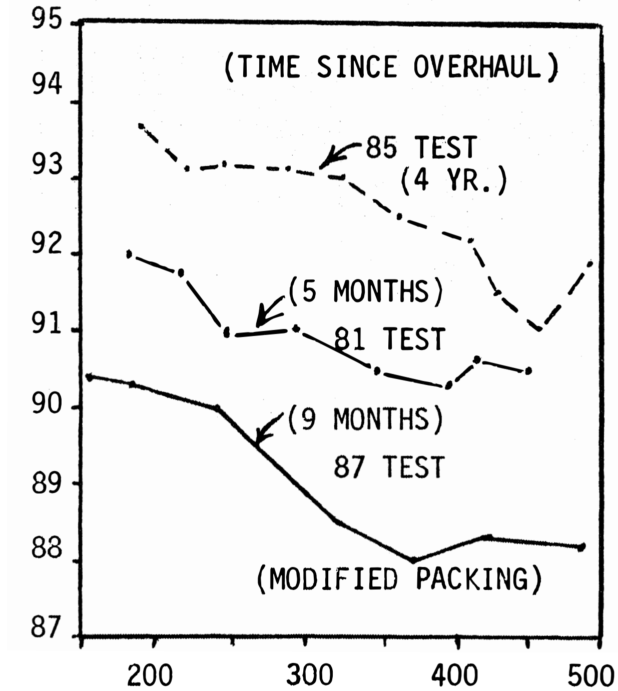

Figure 11. Apparent IP Turbine Efficiency.

Apparent IP efficiency is calculated from the hot reheat to the crossover state points and includes the affect of the relatively cold N-2 packing leakage on the turbine efficiency. Three IP efficiency curves are compared in Figure 11. The post overhaul 1981 test shows the level obtained after five months of operation, while the 1985 data was obtained four years later. The rise in the efficiency during this period indicates the amount of N-2 packing leakage had significantly increased. The 1987 data obtained after the packing modification was installed, indicates the leakage is significantly lower than the 1981 results. This is a good indicator that the N-2 packing clearances are tighter than the 1981 overhaul clearances.

FIRST STAGE HP SHELL PRESSURE

First stage shell pressure will normally remain constant for a given throttle flowrate. If the pressure changes, it indicates a change in flow passing area, either the second stage diaphragm, diaphragm packing or the N-2 packing. This pressure has been trended since the packing modification and shows no sign of degradation in the 12 months of operation, further it is equal to manufacturer design value which strongly suggests design value bypass leakage. Previous testing showed the first stage pressure was 6.5 percent low.

N-2 PACKING RATIO MONITOR

A somewhat unique approach to monitoring the N-2 packing determines the ratio of clearances before the blowdown divided the clearances after the blowdown. This requires pressures and temperature measurements in the blowdown line.

A plot of this data is presented in Figure 12 and step changes are shown at low load that could be a result of the packing opening. A more thorough investigation revealed that the clearance ratio change is correlated to the temperature in the blowdown pipe which drops off significantly at low loads. (Figure 13) This phenomenon is a result of temperature differences between the first stage shell and the IP bowl which results in inner shell distortion. The first stage temperature changes significantly at low loads. The effect has been seen on units with packing boxes and with integral inner shell packing holders similar to Dickerson. This problem is not unique to the modified packing design.

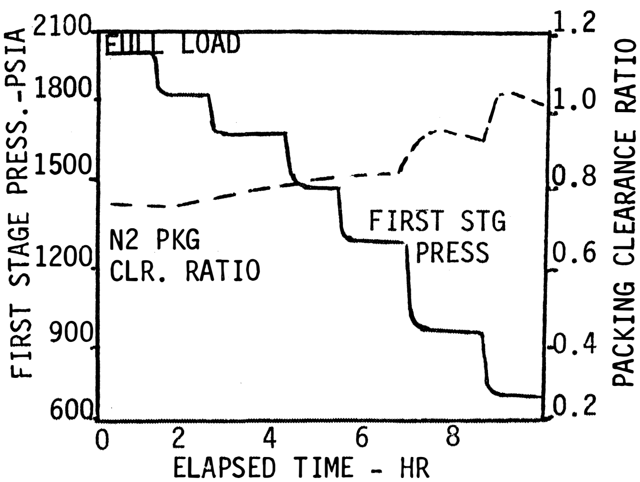

Figure 12. Packing Clearance Ratio.

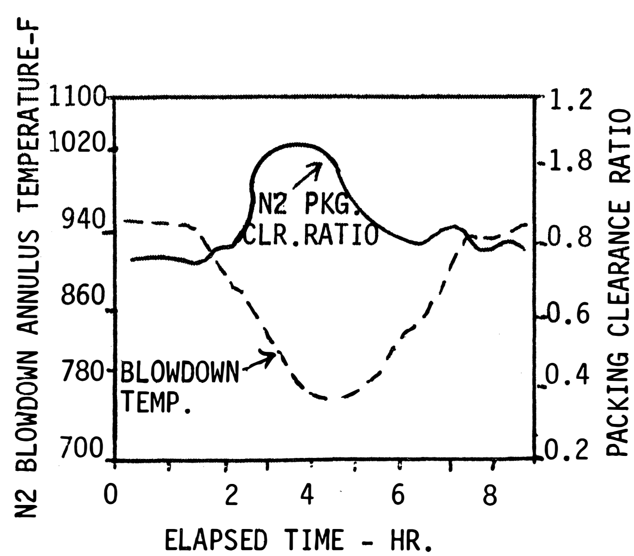

Figure 13. Temperature/Clearance Ratio.

STARTUP MONITOR

The N-2 packing ratio was monitored during startup to observe the pressure changes that would indicate the packing is closing. The expected results should be six unique steps as a result of each of the six N-2 packing rings closing. The data in Figure 14 were taken during a start-up of Dickerson Unit No. 3. The changes in packing ratio are indicative that the packing is closing as the unit is loaded. The number of step changes resulting from packing closures is somewhat interpretative but the on-line clearance ratio does indicate that all packings are closed.

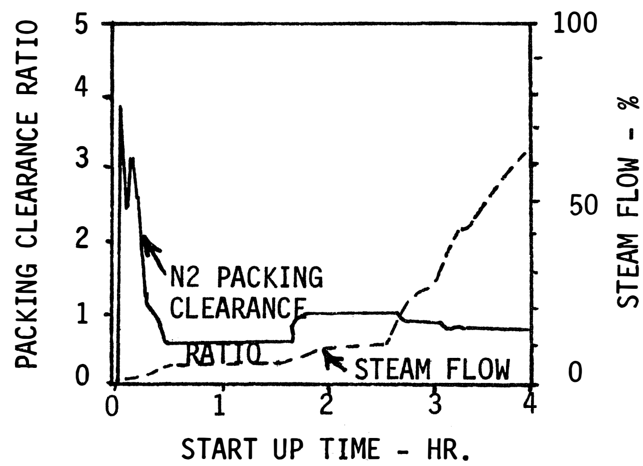

Figure 14. Startup Packing Clearance.

CHALK POINT UNIT 1

The packing modification was installed in Chalk Point Unit 1 during the major overhaul during the Fall 1987. The unit is a supercritical, double reheat, tandem compound turbine. The primary turbine was overhauled and the modified packings were installed in the N-2 packing and the HP turbine and second reheat primary IP turbine diaphragms.

The improvement in pre and post overhaul test of the HP turbine is shown in Figure 15. The gain in efficiency is approximately 7.7 percentage points and results in an improved heat rate of 110 Btu/kWh.

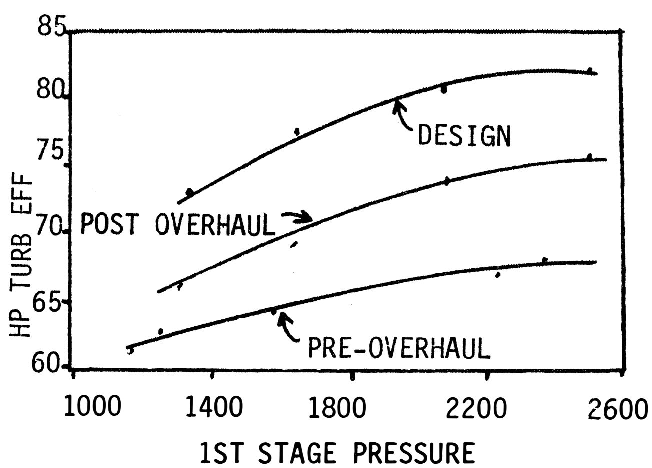

Figure 15. Chalk Point HP Turbine Efficiency.

The primary second IP turbine efficiency improvement is shown in Figure 16. The improvement in actual efficiency of 5.5 percentage points results in a 53 Btu/kWh heat rate improvement. The difference in actual and apparent efficiency of 11.5 percent prior to the overhaul and 3.5 percent after the overhaul shows a dramatic decrease in the N-2 packing flow. The estimated packing flows at full load are 160,000 lb/hr prior to the overhaul and 48,000 lb/hr after the overhaul. This reduction in packing flow results in a 120 Btu/kWh improvement in heat rate.

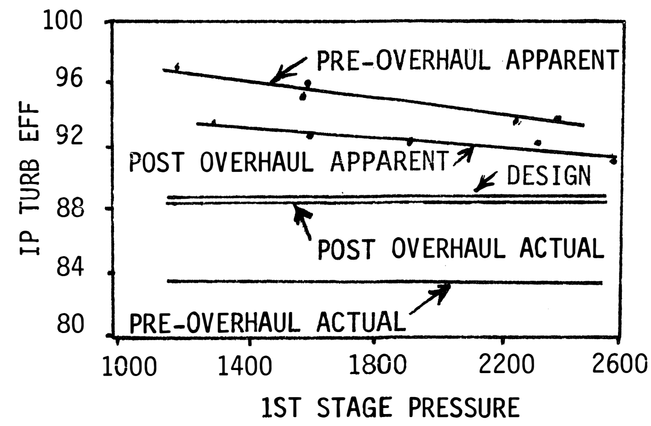

Figure 16. Primary Second IP Turbine Efficiency.

The overall improvement in performance at Chalk Point Unit 1 is 283 Btu/kWh. Based on the overhaul inspection of repairs, approximately 240 Btu/kWh is due to the reduction of tip seal and packing leakage losses. The remainder is the result of steam path repairs.

CONCLUSION

The results show the new packings have improved the performance of Dickerson Unit 3 and Chalk Point Unit 1 significantly. The modification cost was minor compared to the expected gains in performance realized on these two units. If the packings continue to perform over its overhaul cycle as it has to date, then the pay back in fuel savings will be significant.

ACKNOWLEDGEMENTS

The testing, data analysis and presentation of the results for Dickerson was developed by Mr. Greg Staggers, a Project Engineer with Potomac Electric Power Company. The testing, data analysis and results for Chalk Point were the work Dave Schnetzler, a Test Engineer with Potomac Electric Power Company.