A Turbine Cycle Analysis Procedure Used at Potomac Electric Power Company

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ABSTRACT

The goals of turbine testing should be:

1) Document heat rate
2) Develop incremental heat rate curves
3) Use results to optimize heat rate
4) Determine turbine maintenance condition

This paper will review the method that Potomac Electric Power Company (PEPCo) uses to obtain the above goals. These steps include data interpretation and correction, the method used to employ PEPSE in the process, and the display and analysis of the results which lead to diagnosing turbine maintenance condition.

The diagnostic tool, which is reviewed in this paper and given as an appendix, is a procedure (1) written by Ronald E. Brandon of Power Technologies, Inc. for the EPRI Project RP 1681/2153. This project on power plant instrumentation systems is being hosted by PEPCo's Morgantown Unit 2.

INTRODUCTION

The intent of this paper is to review the procedure that Potomac Electric Power Company, PEPCo, is using to analyze turbine cycle test data and to review the methods that have been developed by Ronald E. Brandon for the Electric Power Research Institute, EPRI, to diagnose the turbine maintenance condition using these test results.

The procedure to analyze the turbine test data includes the correct interpretation of the data, the corrections to extraction pressures for pipe pressure drops, and the correct usage of extraction enthalpies for various calculations. Once the pressures and enthalpies have been properly calculated they are input into PEPSE model, along with an accurately measured condensate or feedwater flow and various other plant instrumented flows such as boiler feed pump turbine flows. A control or option to converge wet LP turbine stage efficiency on measured gross generation is also used in the initial test runs for the PEPSE model.

Once the initial results have been obtained from the PEPSE output various plots are made to determine the reasonability and accuracy of the test data and assumptions made. Various calculations are also made to determine such things as N2 packing flow on GE units with the HP-IP turbines under one shell or Hp and IP packing dummy flows on similar Westinghouse designed units. These changes, if needed, are then re-entered back into the PEPSE model to result in the final PEPSE runs.

The results of the turbine stage group efficiencies and turbine bowl flow factors (coefficients) are then averaged, except for the inlet stage of the HP turbine and the wet stages of the LP turbine, and re-entered back into the PEPSE model. Feedwater heater drain cooler approach temperatures and terminal temperature differences are either averaged or scheduled as a function of flow and also re-entered into the modified PEPSE model. The actual test point first stage HP turbine efficiency, main steam flow factor, last stage LP turbine flow factor and wet LP stage group efficiency are also input into the modified PEPSE model. The gross generation control or option is turned off and PEPSE is re-run allowing the megawatts to be calculated directly from all the input data.

Average-run-cases are made to smooth out the individual test case anomolies and to coincide better with the off-design PEPSE runs. Design data is not normally found in the form of design pressure and temperature for a given flow to the turbine, so the off-design runs are made by inputting design section efficiencies and design flow factors for the particular stages in question. The internal design calculation in PEPSE can also be used if so desired.

Once the results are obtained they are displayed in a form which is useful to diagnose turbine maintenance condition. The procedure (1) developed by Ronald E. Brandon of Power Technologies, Inc. (PTI) for EPRI can then be used to diagnose turbine maintenance condition. This procedure consists of nine steps that have been used for years to diagnose the internal condition of the turbine, but have never been combined in one single paper.

ANALYSIS REQUIREMENTS

The first requirement to a turbine cycle analysis is an accurate set of data taken at steady state and isolated unit conditions. Probably the most important factor is cycle isolation. Isolation means that the cycle is set up as close to design cycle heat balance as possible. If differences exist, measure their corresponding in or out flows to the cycle. As long as the deviate flows are known, they can be taken into account in the turbine cycle analysis. Sufficient data should be taken to calculate a turbine cycle flow and energy balance. This data should include but not be limited to:

(1) Pressure and temperatures of all dry steam state points.
(2) Pressure of all wet steam state points.
(3) Pressure and temperatures of all feedwater heaters (FW in and out, extraction T&P, drain temperature).
(4) An accurate flow measurement, usually either condensate flow or feedwater flow. (See Figure 4.1 of ASME PTC 6S).
(5) Gross generation and pertinent generator data.
(6) Turbine control-valve position data relative to flow.

The focus of this paper is on analysis rather than data acquisition so accurate data is needed across the load range at steady state/isolated unit conditions. An excellent paper written by Hopson, Peyton and Legg (2) on cycle isolation is recommended reading before isolating a unit for testing.

FLOW AND ENERGY BALANCE DATA PREPARATION

Pressure Corrections

One of the first tasks of analysis is to validate the test data and prepare it for the particular calculation. The test engineer should be careful and know where the temperature and pressure measuring points are physically located in the piping. For extraction enthalpy calculations, where pressure and temperature are measured at substantially different locations, the pressure must be corrected for pressure drop between the pressure tap and the thermowell.

Enthalpy is assumed constant in most piping runs but temperatures change with pressure drop. Pressure should also be corrected for pressure drop from the point of measurement to inside the turbine stage exit or to the heater shell neck for the particular application. For example, do not use the extraction pressure measured and corrected to the heater neck for expansion line plots or flow factor (W/(P/V)**0.5) calculations. The pressure should be corrected for pressure drop to the turbine stage exit condition. Omitting this correction can lead to significant errors in the flow factor calculation. Architectural engineering heat balances are good for estimates of pressure drops in the piping. If they are unavailable the turbine manufacturer's pressure drops in the extraction lines should be used. The Spencer, Cotton and Cannon paper (3) uses a three percent drop from the stage exit to the turbine flange and another three percent from the turbine flange to the heater.

Enthalpy Corrections

Steam temperature measured in extraction piping is normally hotter than the average steam temperature inside the exit of the turbine stage. Some steam in the turbine stage bypasses the blade partition and goes through the shaft packing or spill strip, and does no work. Therefore it is hotter than the steam which passes through the partition and does work. Because this by-passed steam flows around the tip of the blade, some of it is diverted down the extraction pipe to the heater. Sometimes the difference is as high as 15 to 20 BTU/lb. What is needed for the turbine/generator energy balance, flow factor calculation, and the efficiency calculation is the enthalpy of the steam inside the turbine stage. On extractions at turbine section exits, such as cold reheat or crossover, use of the measured enthalpy is correct.

To alleviate the hotter than average steam extraction problem, enthalpies should be read from an expansion line drawn for the HP and IP/LP turbines. A straight line is used for the HP turbine expansion line. For the IP/LP turbine on reheat units, a Keuffel and Esser curve #1864-31 can be used. The expansion line is drawn from the IP bowl condition through the crossover condition using the corrected pressure and enthalpy. (Note, if the expansion line end point is known, use this instead of the crossover point to draw the expansion line.) Non-reheat units should be drawn with a Keuffel and Esser curve #1864-41. Use of the Keuffel and Esser curves are also referenced in other papers such as the Hegetschweiler and Bartlett paper (4), and the PEPSE manual user description (5).

Once the expansion lines are drawn, the enthalpy of the extraction steam is determined by where the extraction pressure, corrected to the turbine stage exit pressure, crosses the expansion line. Figure 1 shows such an expansion line for a turbine and the actual measured data in the extraction pipe. The enthalpy on the expansion line should be used with the extraction pressure for any turbine calculation such as flow factor, efficiency, or the turbine/generator energy balance.

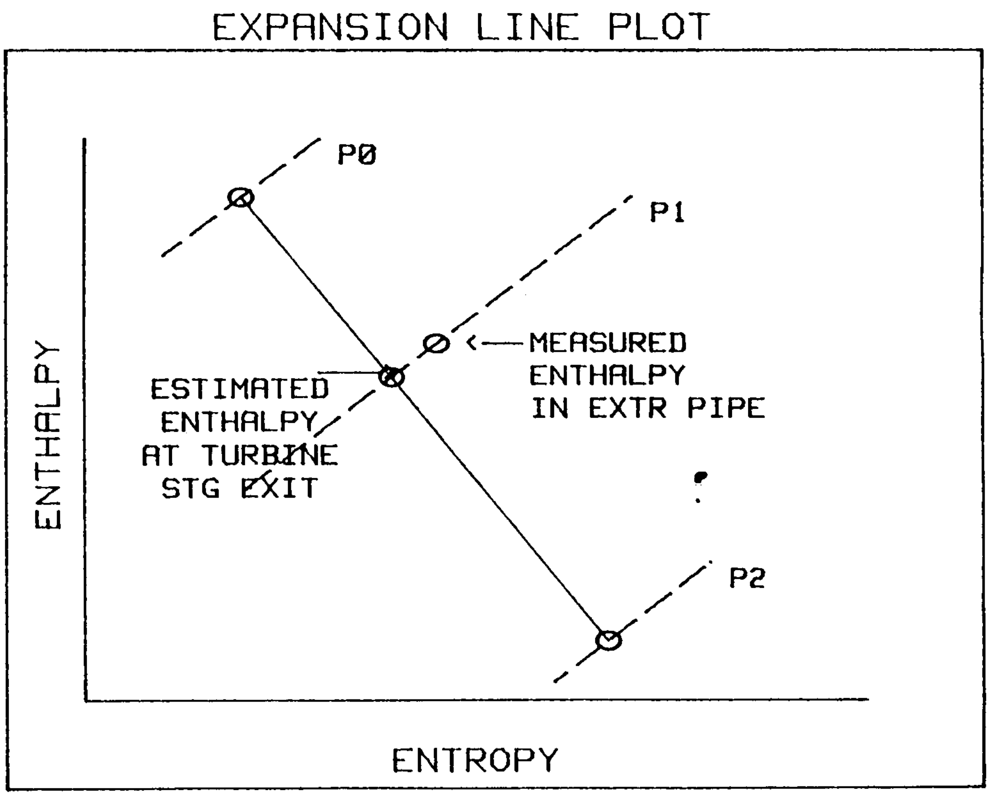

Figure 1. Expansion Line Plot

To obtain an even more accurate result an energy balance and a mass balance should be done for each extraction. The reason being, the average extraction steam is hotter than the average steam exiting the turbine blade, which is hotter than the average steam entering the following stage. There are three different enthalpies involved in each extraction (unless the extraction occurs at a turbine section exit such as cold reheat or crossover). Figure 2 illustrates this concept in block diagram form. Again, the reason that H_2 is hotter is due to spill strip leakage which is diverted down the extraction pipe.

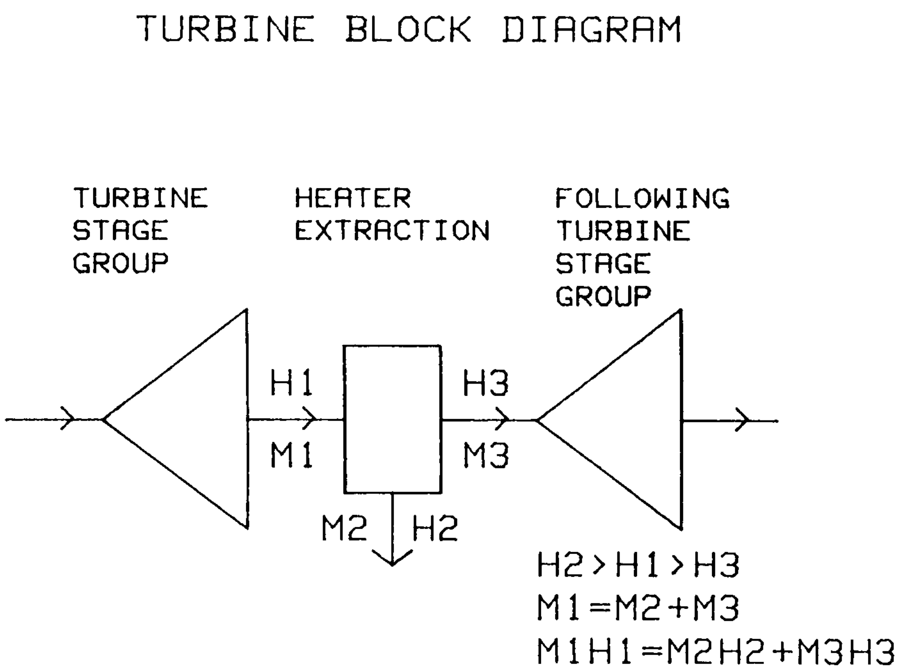

Figure 2. Turbine Block Diagram

H_2 is measured and all flows are relatively known, so the only unknowns are H_1 and H_3. Figure 3 represents the corresponding expansion line of this extraction. This type of calculation tends to be a lengthy one, when considering that the process of the energy balance on the cycle is an iterative one, and is usually done on a computer.

H_2 should be used in the feedwater heater energy balance calculations, assuming there are no other flows mixing with the extraction before it reaches the heater. H_1 and H_3 should be used for the energy balance between the turbine work and the generator-output-plus-losses calculation usually performed to solve for the wet turbine extraction and the expansion line end point. H_3 should also be used to obtain specific volume for the bowl flow factor calculation. The difference between H_1 and H_3 depends on the amount of spill strip leakage, and the flow through the turbine at that particular stage in question. Normally this difference is very small.

EXPANSION LINE PLOT

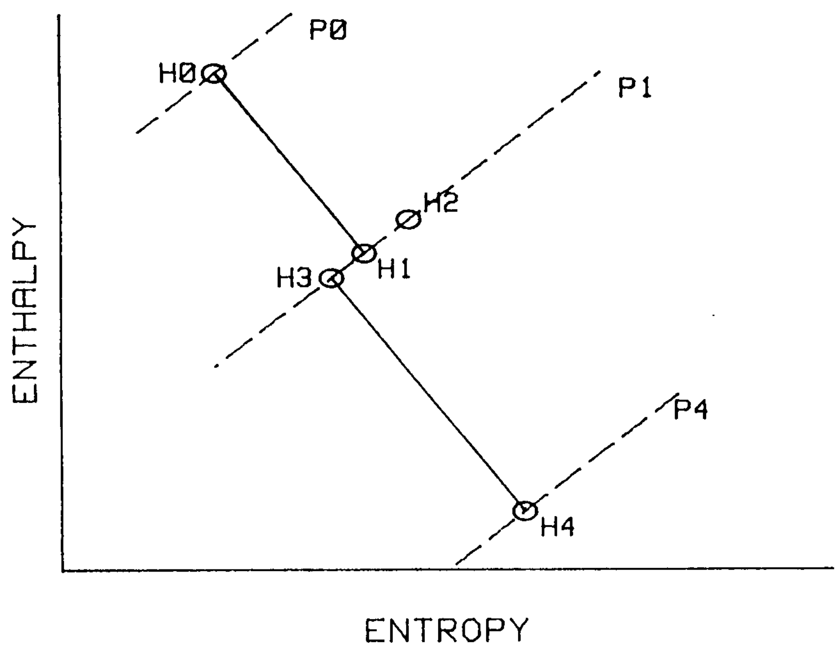

Figure 3. Expansion Line Plot

Initial PEPSE Runs

Once the initial corrections have been made to the data set the data is input into PEPSE. Correct usage of extraction pressure and enthalpies and estimated extraction line pressure drops, as discussed earlier, should be input into the PEPSE model.

Normally, a calibrated flow nozzle section, including a flow straighter, is installed in the condensate line or feedwater-to-economizer line to accurately measure flow. From this measured flow nearly all other flows including extractions can be derived. This flow will be used in conjunction with a control to obtain main steam flow in the model.

Some inputs to the model will be estimations or measured flows using existing uncalibrated plant flow elements. Included in these estimations are turbine shaft packing flows and enthalpies such as N2 flow and enthalpy on GE units whose design incorporates the HP and IP turbines under one outer shell or HP and IP dummy flows and enthalpies on Westinghouse units of similar design. Boiler feed pump turbine steam flows and superheater and reheater spray flows are usually inputs measured with unknown physical condition flow elements whose accuracies should be questioned.

A control or option that swings the wet LP stage turbine efficiency to balance the turbine energy with generator-output-plus-losses should also be input to the model.

After the PEPSE cases are run on the data that was taken across the load range the results are then tabulated and various plots and calculations are made to determine the data reasonability and accuracy.

One such calculation is that of IP turbine efficiency and the accounting of the N2 flow which mixes in the IP bowl. If the N2 flow (or IP dummy flows in the Westinghouse case) are not known accurately then reheater duty will be inaccurate and IP and LP turbine efficiencies will be in error as well as turbine cycle heat rate. An accurate account of N2 flow also helps to estimate any other leakage flows which might occur such as steam lead snout ring (GE) or bell seal (W) leakage flows.

There are several methods for calculating N2 flow without actually measuring a flow. One such method is described in a paper by GE.(6). Another method employs the IP turbine efficiency, which is known to be a constant over the load range. If the N2 flow that affects the reheat bowl enthalpy mix is incorrect, the efficiency over the load range will be too high and will slope from high efficiency at low load, to lower efficiency at high loads. Multiples of N2 flow are iterated to correct the bowl enthalpy until the efficiency over the load range is horizontal. Figure 4 depicts this situation. In many cases an exact horizontal line can not be obtained so engineering judgment must be used.

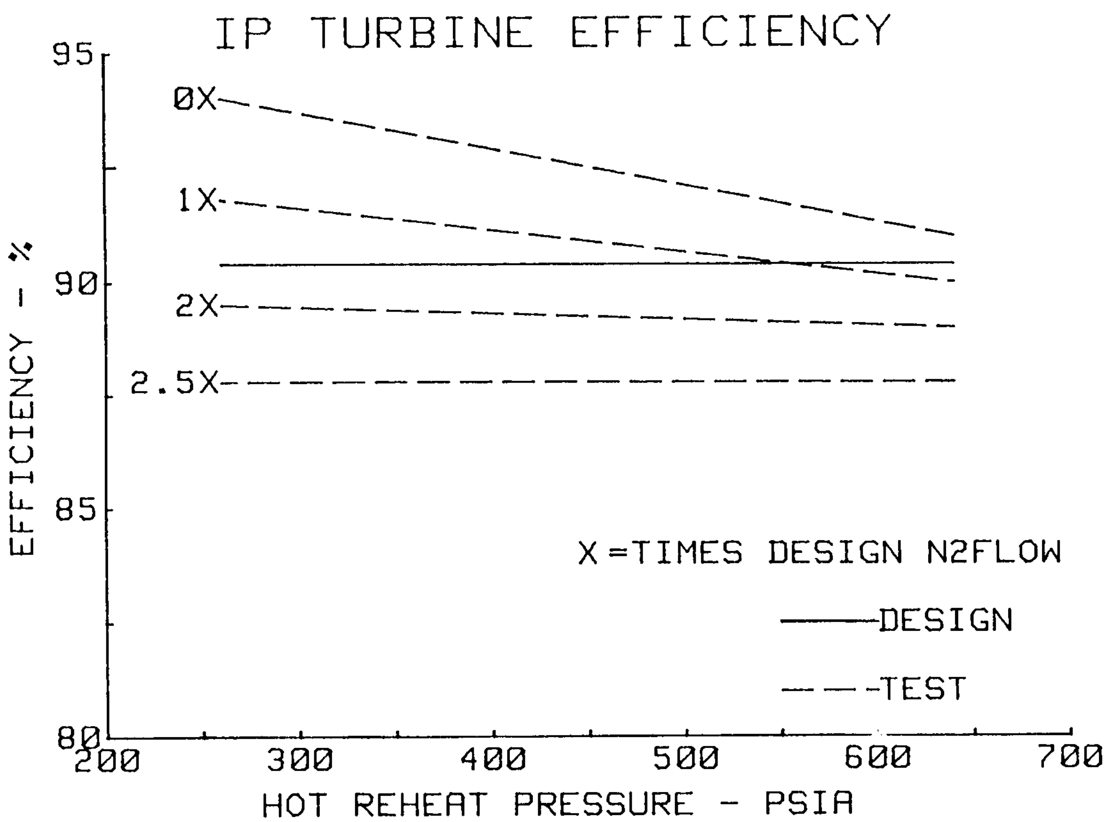

Figure 4. IP Turbine Efficiency

One other method that is not absolute but relative is to use the "apparent" IP section efficiency at a known N2 packing clearance (like after overhaul) and compare this to the apparent IP efficiency at some later date. Figure 5 depicts this situation. If no IP turbine efficiency degradation is assumed, the difference in the efficiency must be caused by increase of N2 flow. Apparent efficiency is based on the conditions at the hot reheat before the N2 flow mixes in.

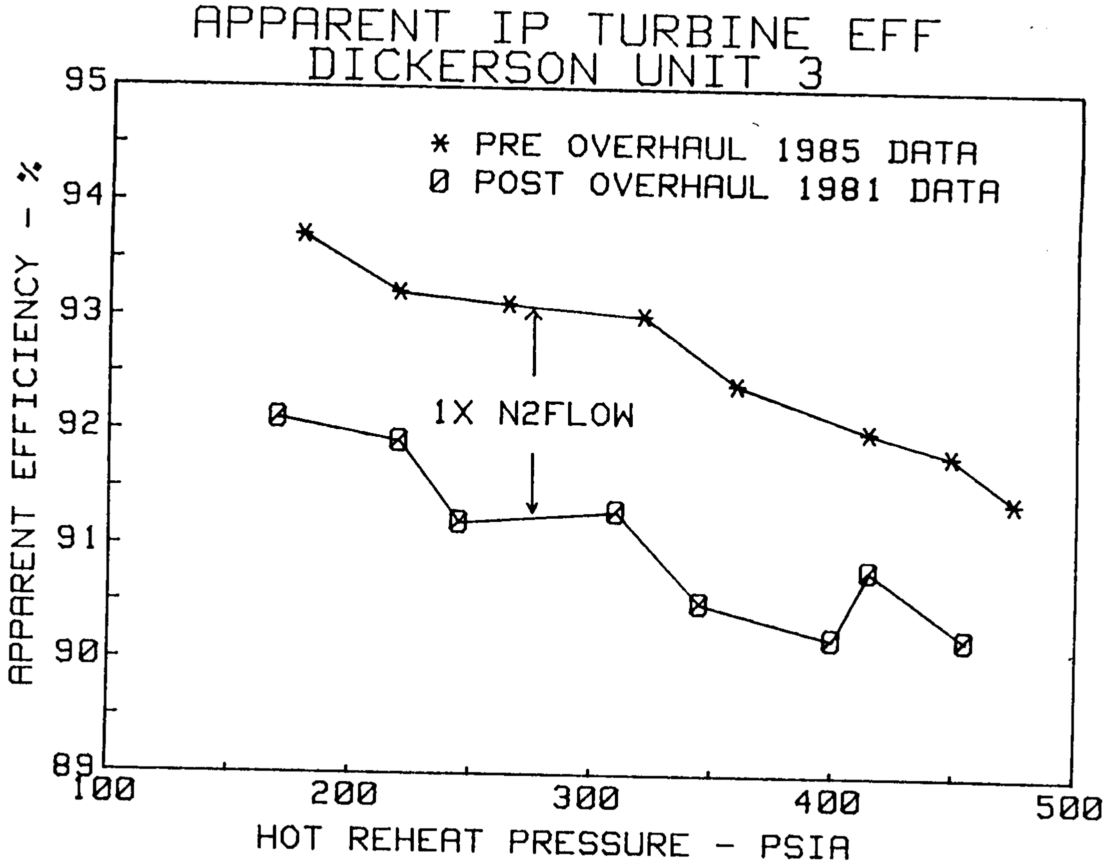

Figure 5. Apparent IP Efficiency

Success with these methods depends on the sensitivity of the N2 enthalpy on the IP bowl enthalpy. If N2 enthalpy is nearly equal to the hot reheat enthalpy, the calculation would be very insensitive to N2 flow and large errors could occur. Also the N2 enthalpy is, in many cases, a "calculated value" and not very accurate.

The best way to obtain an N2 flow is to use a blowdown system like the one used on Morgantown Unit 2, the host unit to the EPRI project. These systems are discussed in more detail in the Brandon paper (1) and the Booth and Kautzmann paper (6).

Flow Factors

The PEPSE calculated flows through the turbine are used to calculate flow factors, (W/(P/V)**0.5) and are compared to design heat balance or some baseline test data. Flow factor is a coefficient that is proportional to flow passing area of the following stage diaphragm. Figure 6 depicts the flow factor measurement points.

Flow factors are used to validate data and to indicate diaphragm flow passing capability. Flow factors plotted across the load range should normally be linear and horizontal lines for all stages except the first stage nozzle (using main steam flow, pressure and specific volume) and the last stage of the LP turbine. If the line is not horizontal but tilted or data are

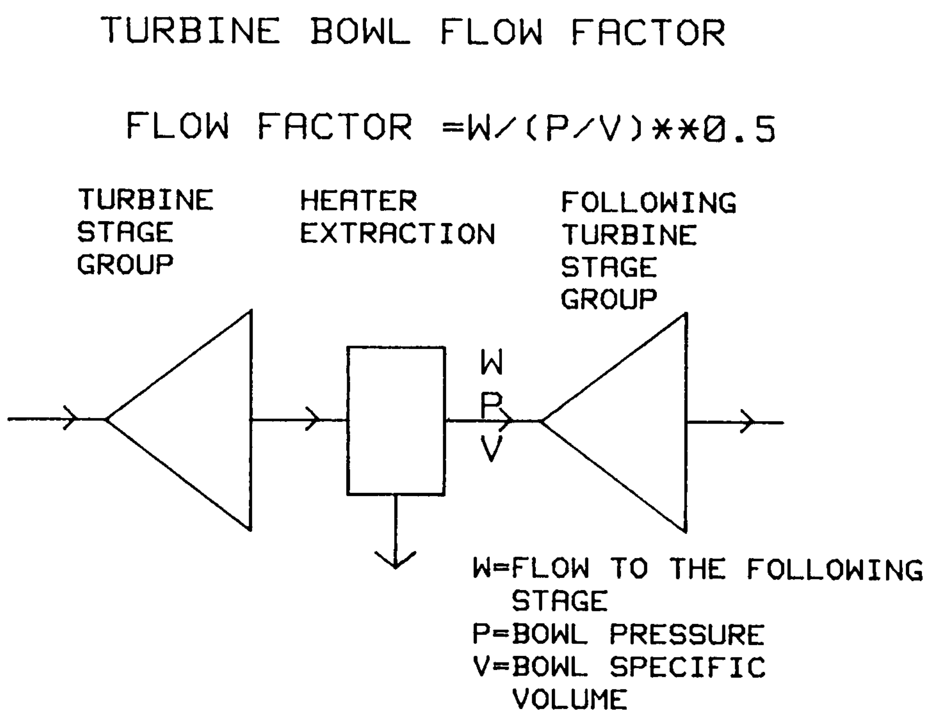

Figure 6. Turbine Bowl Flow Factor

scattered, it could mean that the flow, pressure or specific volume are incorrect. Figure 7 depicts this situation.

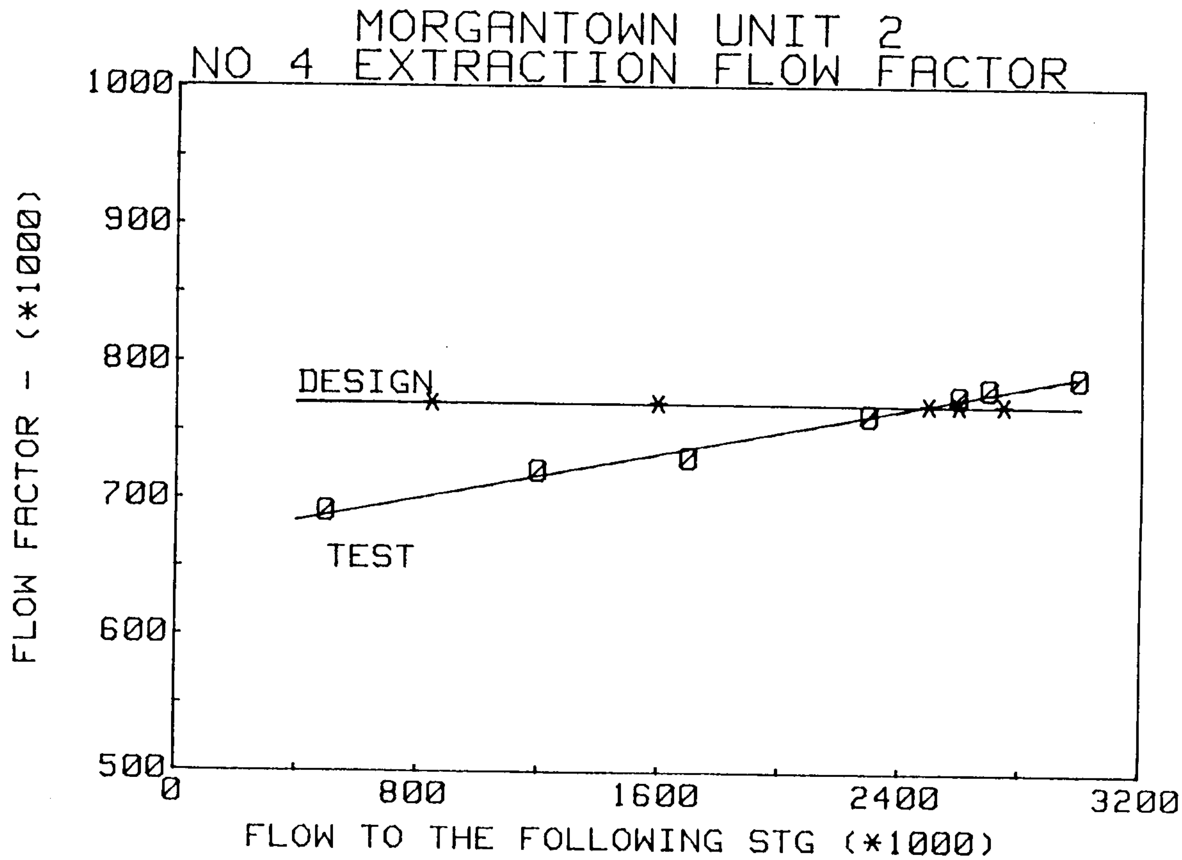

Figure 7. Extraction Flow Factor

If pressures and temperatures are verified then suspect flow. The engineer should determine whether flow could bypass the stage or enter the stage at some point to affect the calculated flow. For example, are there other heat/flow sources going to the heater that are not being taken into consideration and cause errors in the calculated flow? Turbine cross-sectional drawings must be examined to analyze different leakage paths such as snout rings or bell seals on main steam leads.

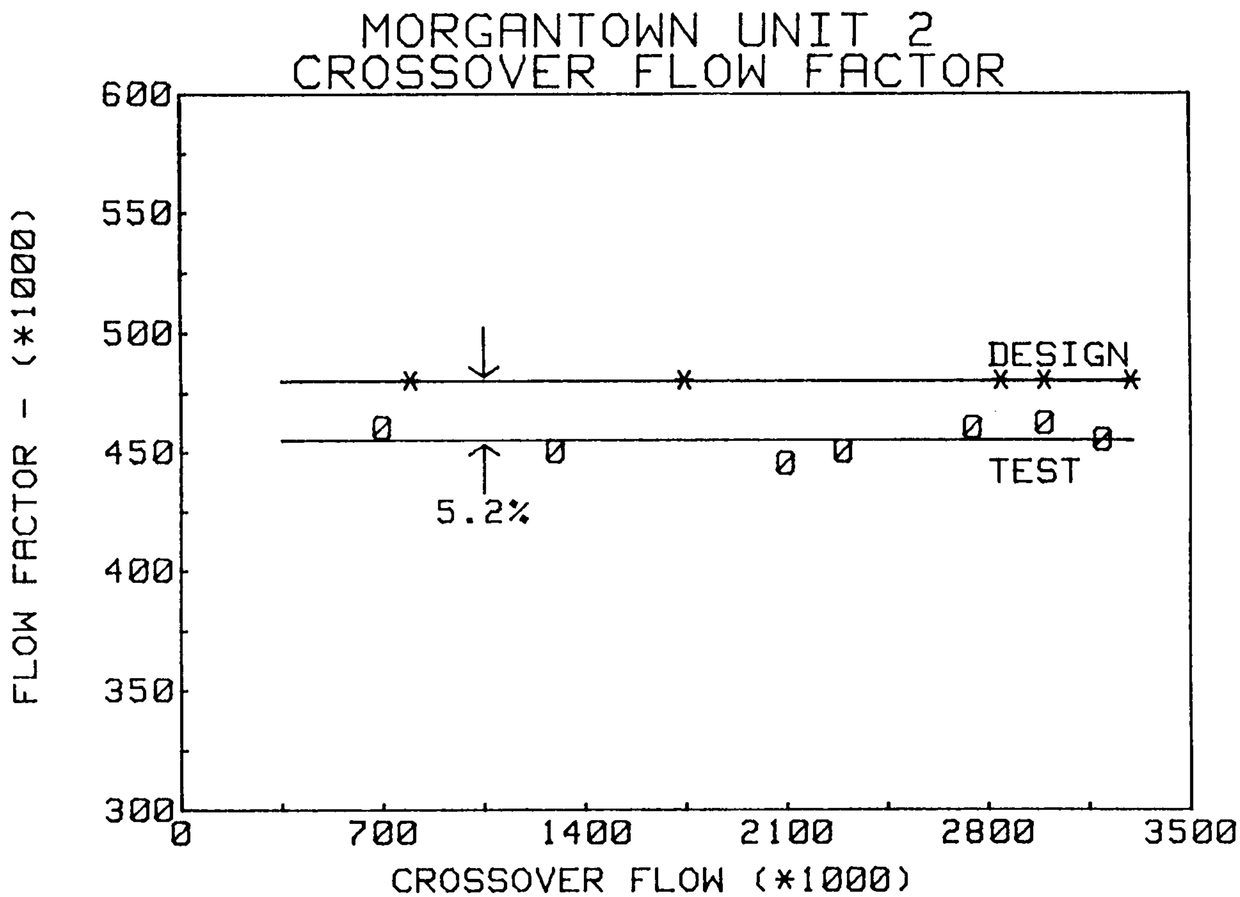

Figure 8. Crossover Flow Factor

If the flow factor plot is horizontal but at a different level than previous tests or design (Figure 8) it could indicate an area change or maybe a flow error across the load range. If a flow error is suspected other flow factors should be checked at different stages in the turbine that are believed to be near design. Normally, an IP turbine extraction or crossover point tends to be a good comparison point to design, since these stages are rarely eroded or damaged. The pressure and enthalpy must be corrected values, as discussed previously. If the IP extraction or the crossover sections also show a different than design level of flow factor and if the difference is about the same percentage as the suspected flow factor, flows are probably incorrect. The problem could also be that the main flow measurement is in error, or that, flow is bypassing the turbine.

Another validation of the data is the result of the turbine/generator energy balance. The energy balance solves for the wet turbine stage extraction enthalpies, the expansion line end point (elep) and the used energy end point (ueep); in other words, low pressure turbine and reheat turbine efficiency. These efficiencies should be compared with design or expected values. If they appear unreasonable there is a problem with data or something has not been taken into account. In the Brandon procedure (1) the KW check performs a similar task.

If the difference between test and design flow factor is attributed to flow passing area the correlation is not one to one. A curve is given in the Brandon paper that, given the percentage change in flow factor and the pressure ratio on the stage, yields the percent change in area.

PEPSE Second Runs

Now that the initial results have been reviewed, any problems found with flow or any other input data that results in an erroneous output, should be changed in the PEPSE model and re-run. Once these are done, final plots of the efficiency, heat rate, flow factors, etc. can be made.

These results will be used to document the as tested unit condition. From these results two of the four goals listed in the abstract can be accomplished, they are:

- Document heat rate
- Provide the input needed for the incremental heat rate curves.

PEPSE Average Runs

Since most turbine group efficiencies and flow factors are constant throughout the load range and some test points may have their own data anomolies, it is best to average the PEPSE results using all the test points across the load range and re-run PEPSE using the average data. Another reason to average the results and re-run PEPSE is design data is given in the form of flow factors and efficiencies and not pressures and enthalpies.

For example, a unit was tested a year before its scheduled maintenance overhaul and it is desired to know what effect a rebuilt IP turbine will have on unit performance. The turbine is assumed to be rebuilt to design conditions. Design efficiency for the IP turbine and design flow factors for each stage group in the IP turbine are input into the model. If the HP turbine inputs were left as test pressures and temperatures the PEPSE results would be in error. The reason being, for a given flow to the HP turbine (i.e. constant control valve position) the reheat bowl pressure, and hence the hot reheat pressure, would change due to the new design inlet area of the IP turbine and therefore the cold reheat pressure would change. Therefore the entire original PEPSE model must be changed to account for flow factor and efficiency inputs instead of pressure and enthalpy inputs.

As stated earlier most turbine efficiencies and flow factors are constant throughout the load range. The exceptions are the first stage HP turbine efficiency, main steam flow factor (which is set by control valve position), wet LP turbine stage efficiencies and the last LP stage flow factor. These can not be averaged and must be specifically input to the model for each test point run.

Other data that can be averaged or scheduled as the case may be are feedwater heater drain cooler approach temperatures and terminal temperature differences.

These data are then input back into a modified PEPSE model which will take these types of inputs. Since the main steam flow factor is input into the model the control in the initial model that changes main steam flow to obtain the given measured flow (condensate or feedwater-to-economizer) is no longer needed. A new control or option that changes main steam flow to obtain the given main steam flow factor is used. The control or option which swings the LP turbine efficiency is no longer needed either since the wet LP turbine stage group efficiency is directly input into the model. The wet LP turbine stage group efficiency input into the model should correspond to the correct input back pressure, otherwise errors will result.

PEPSE Design Runs

The average-data-model-results of heat rate and generation for each test point are used as a benchmark to compare to the results of the design PEPSE runs.

The difference in heat rate and generation between the average test model and the design runs, are then applied to the individual test heat rate and generation that was obtained in the earlier runs.

Normally the comparison of design runs to test runs are made at constant control valve position. This implies that the main steam flow factor is the same for the test and design runs. The individual test run's main steam flow factor is input into the design run model to assure the valves remain at a constant position.

PEPCo normally make design runs for the HP, IP and LP turbine sections, design packing for the N2 or HP-IP dummy, and all design GE-2 corrections. Combinations of these are run to find the effect each has on heat rate and generation. Various other runs for best achievable data are also run.

The design run heat rate and generation when compared to the average test run heat rate and generation, result in the difference the design change has on the cycle.

This data can be used to economically justify ordering new or rebuilt parts for upcoming turbine overhauls. Cost benefits can be done on these results too that could change the overhaul schedule. This is a cost savings in itself.

Brandon Paper Review

After the test analysis has been completed the turbine can be diagnosed for maintenance condition by using the individual-test-data-results in conjunction with the Brandon Procedure. The Brandon paper which is attached as an appendix to this paper, is a formalized guide on interpreting the results of the turbine cycle heat and energy balance to predict turbine maintenance condition. The analysis procedures described in this paper have existed for years but have never been fully documented in one paper till this time. This procedure is a product of the EPRI project RP 1681/2153 which is ongoing at PEPCo's Morgantown Unit 2.

Reference (1) outlines nine steps used to interpret turbine cycle test data:

(1) Flow capacity check.
(2) Throttle flow factor across the load range.
(3) VWO flow/1st stage pressure analysis.
(4) Flow factor analysis at hot reheat, xover and heater extractions.
(5) HP turbine efficiency across the load range.
(6) IP turbine efficiency across the load range.
(7) VWO kilowatt check.
(8) Other considerations (Isolation, Leaks, etc.)
(9) Probable causes of section loss.

Step 1 of the procedure calls for checking the turbine throttle flow at valves wide open (VWO) condition with design heat balance. The test throttle flow is corrected for main steam temperature and pressure. The design flow for G.E. unit, is normally 2.5 percent greater than VWO heat balance flow. This step can result in the prediction of the following:

- First stage nozzle erosion;
- Bypassing leakage flows not taken into account;
- Excessive N2 or second stage packing clearances, or
- A change in second stage diaphragm flow passing area.

Step 2 of the procedure is similar to Step 1, but uses the turbine throttle flow factor and makes the comparison across the load range. If the test is done at valve-crack-points, and valve-crack-point heat balances or baseline test data taken at valve-crack-points are available, the condition of each nozzle box arc's flow passing area can be determined and compared to design. Figure 9 depicts this situation. The throttle flow factor is corrected for main steam pressure and temperature. Control-valves must be checked for alignment before testing; otherwise, erroneous data comparisons may result.

The flow passing ability of a nozzle or diaphragm is not only a function of area but also of pressure ratio. If the pressure ratio of the stage in question is greater than acousitc (1.83) then the area change of the following stage diaphragm or nozzle will be directly proportional to flow change. If the pressure ratio is less than acoustic, the area change is a function of restriction factor and flow. Figure 10 shows the relationship between restriction factor and pressure ratio, where A is area, W is flow, and N is restriction factor.

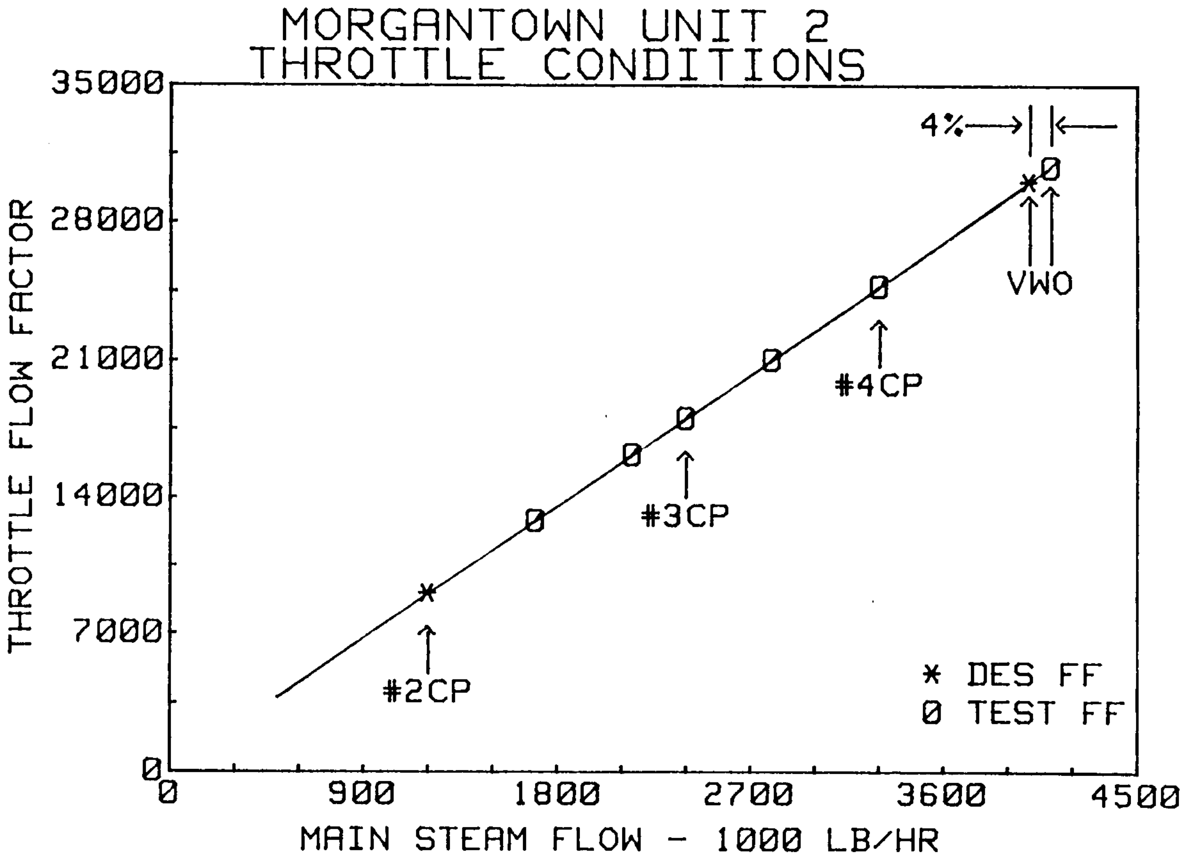

Figure 9. Throttle Flow Plot

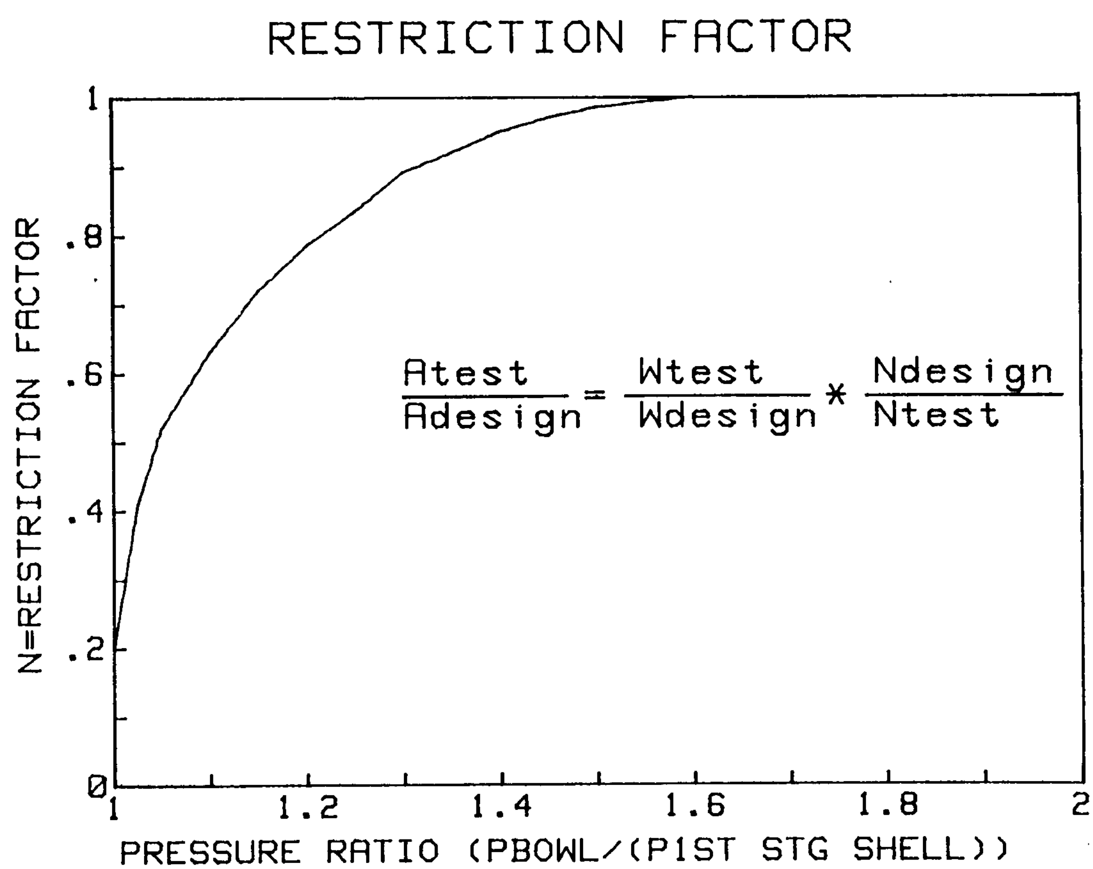

Figure 10. Restriction Factor Plot

Step 3 compares tested with design VWO main steam flow versus first stage shell pressure. Figure 11 shows such a plot. The results of this step can indicate the following:

- Changes in N2 or second stage diaphragm packing clearances;
- A change in second stage diaphragm flow passing area;
- Other bypassed leakage flows not taken into account.

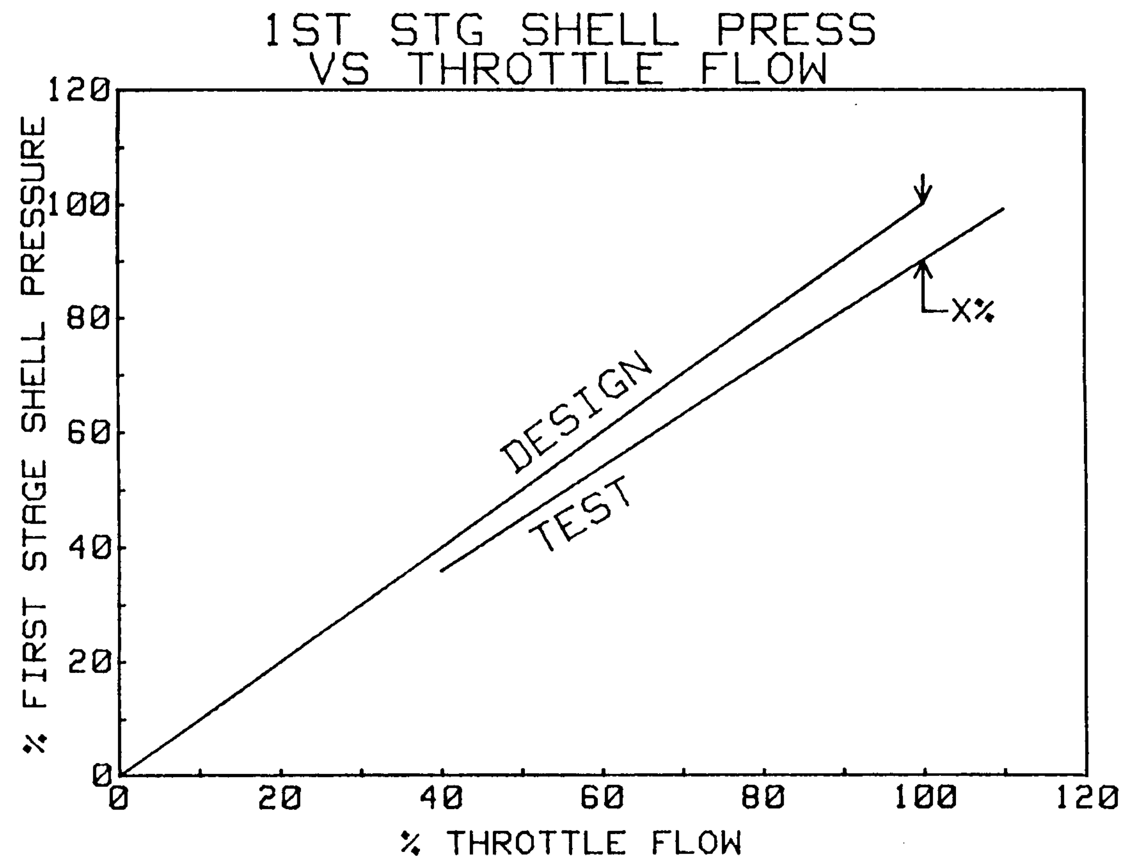

Figure 11. 1st Stage Shell Pressure Plot

Step 4 is a flow factor analysis for hot reheat bowl, crossover and extraction points over the load range, which can be used in three ways:

(1) Estimate diaphragm flow passing area changes.
(2) Deduce flow by using the turbine diaphragm stage as a flow nozzle.
(3) Validate test data.

The corrected pressure and enthalpy must be used at the following stage entrance to calculate flow factor, as discussed previously.

Step 5 compares test and expected HP turbine efficiency across the load range. Not only can the efficiency curve indicate the section efficiency, but also it can inform the engineer whether the first stage or latter stages are damaged. Figure 12 indicates that the first stage is more damaged than the latter stages. This is due to the first stage doing a greater percentage of the HP turbine work at light loads and therefore its effect on HP <u>section</u> efficiency is more at low loads. The reverse can be said if the high load efficiency degradation is more severe than the low load efficiency degradation.

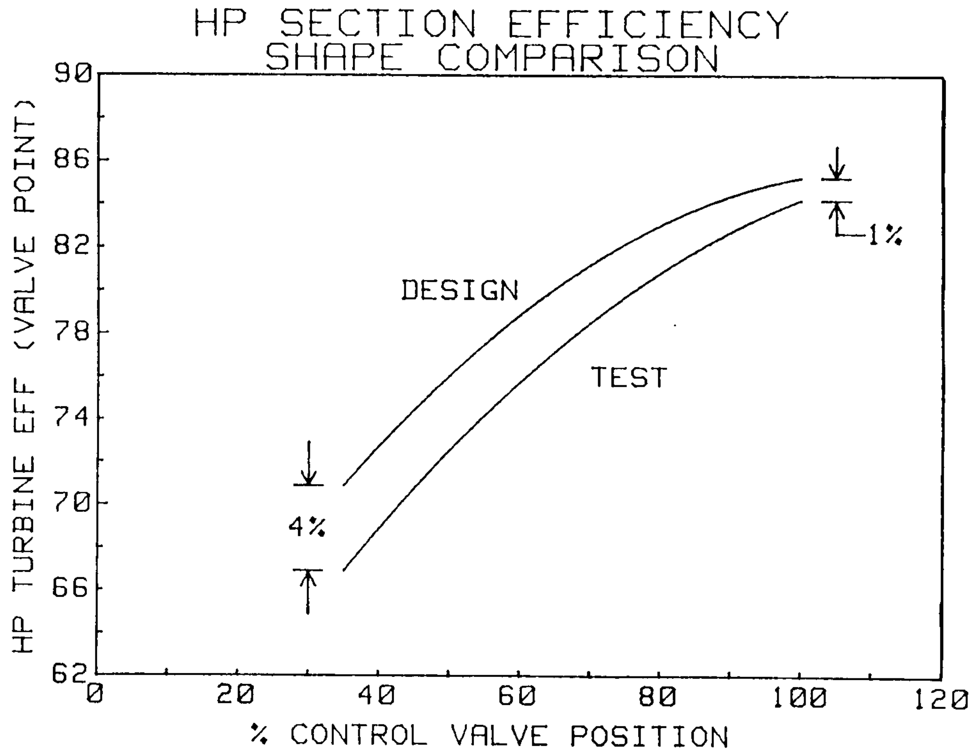

Figure 12. HP Turbine Efficiency Shape

Step 6 compares test and expected IP turbine efficiency across the load range, Figure 13. As stated earlier, IP efficiency is constant over the load range and can be difficult to calculate due to the bowl leakage effect.

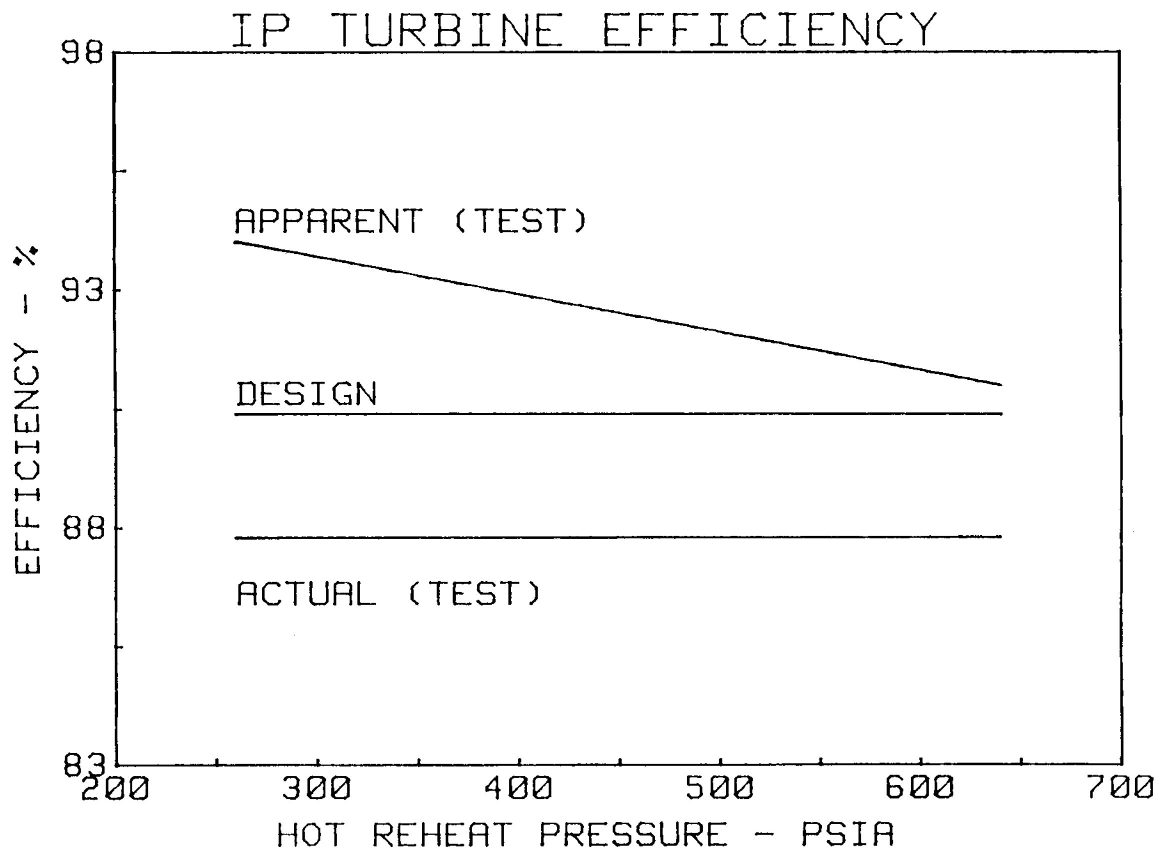

Figure 13. IP Turbine Efficiency

Step 7 of the procedure is a VWO kilowatt check. The kilowatt check is accomplished by correcting test data back to VWO design heat balance conditions and calculating the kilowatt affect for each correction. This step is used to validate test data and analysis assumptions, and to estimate the LP turbine losses. After all corrections have been made, the corrected test kilowatts should equal the design heat balance kilowatts except for the LP turbine losses. This is true if the data are valid and the analysis assumptions are correct. This step is actually already accomplished by the PEPSE run results. But, if an inhouse heat balance model or hand calculation is made, and the wet LP turbine stage enthalpies are not calculated, then this step is mandatory.

Step 8 of the procedure suggests that, if the flow factors of the turbine sections are systematically off design by a similar percent then there is a possibility of some type of leakage flow around or into these stages or the possibility of a flow error exists.

The following are likely places for flow errors to occur:

- o Improper unit isolation (eg. start up drains leaking).
- o Main steam lead leakages (snout rings on GE units and bell seals on W units).
- o Uncalibrated existing plant flow meters on items such as main or reheat spray flow or boiler feed pump turbine flow.

Step 9 determines the probable cause of the turbine section loss. Usually a parameter, such as turbine efficiency is plotted against time to determine whether the change was a gradual one or a step change.

If the change is gradual, erosion or deposits should be suspected. If the change is sudden, leakage flows or other internal damage should be suspected.

PEPSE Use In The EPRI Project

Energy Incorporated has donated the use of their PEPSE computer model to the Electric Power Research Institute (EPRI) for project RP 1681/2153 which is presently being hosted by Potomac Electric Power Company's (PEPCo) Morgantown Unit 2. The overall goal of the project is to develop a state-of-the-art performance-monitoring demonstration system. PEPSE is being utilized in several areas of the project.

Morgantown Unit 2 turbine cycle has been modeled in detail using the PEPSE program. The model is being used as an aid to verify the on-line performance calculations which are continuely running on the system. Periodic turbine tests run on the unit have been analyzed using the model to obtain results needed for documenting heat rate, and obtaining the necessary results used to diagnose turbine maintenance condition.

The PEPSE model is also being used to help verify a boiler computer model called "HEATRT", developed by Lehigh University, one of the main contractors in the boiler section of the project. The code "HEATRT" has been developed to compute the effects of fireside parameters, such as excess oxygen, coal grind size and exit gas temperature on the performance of a pulverized coal unit. Field boiler tests have been run on Morgantown Unit 2 and the data analyzed, with the use of PEPSE for the turbine cycle heat rate, to help verify and tune the "HEATRT" model.

SUMMARY

The product of a turbine cycle analysis is the determination of turbine component conditions and turbine cycle heat rate. The steps leading up to these results involve knowing the data and using the correct data for the particular calculation. After the data are available in a usable form for analysis, predictions can be made on turbine component condition. These predictions enable such steps as the following:

- Advanced scheduling of long lead time parts;
- Possibly change overhaul schedules and scope based on economics, and
- Change operating procedures that adversely effect turbine maintenance condition.

Turbine cycle heat rate and a measured boiler efficiency result in unit heat rate which is needed for updating incremental heat rate curves which the units dispatch on.

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APPENDIX

RP 1681/2153
Power Plant Performance Instrumentation System

PRE-OUTAGE TURBINE EVALUATION AND ANALYSIS PROCEDURE

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Section 7

PRE-OUTAGE TURBINE EVALUATION AND ANALYSIS PROCEDURE

INTRODUCTION

A systematic analysis of historical performance tests and inspection reports can be used to improve maintenance planning. The objective of this paper is to outline one such analysis system used to predict the internal condition of a turbine prior to being opened for inspection.

The evaluation of data from a test prior to shutdown was used to anticipate the condition of the steam path of the turbine, enabling improved planning for spare parts and repair procedures. At a subsequent internal inspection of the turbine, the evaluation was also used for comparison with the actual physical condition of the steam path including calculations of loss due to deterioration of buckets, nozzles, packings and other flow path components.

After completion of repair of the turbine, its condition was again evaluated to determine the expected performance based on the repaired condition. Performance tests after startup and analysis of the associated data was the final calibration step in the process of test-inspect-repair-test evaluation.

The Morgantown 2 unit was used to illustrate the use of the procedure. The evaluation of that unit both before and after the major outage of May 85 was a vital step in the procedure. For reference, a turbine cross-section (HP/IP) and a heat balance are included in Appendix VI.

GOALS

The major purpose of the following analysis techniques is to enable reasonable pre-inspection predictions of the internal steam path condition of large steam turbines.

Such predictions enable improved planning including: selection and ordering of high-cost, long lead time spare parts; probable outage repair procedures; modifications to operating methods; recommendations for improved instrumentation and test activities; optimization of outage timing; and scheduling on a cost/benefit basis.

APPROACH

The evaluation and analysis techniques are presented in four sections as outlined below:

- o Summary of the specific analyses with individual goals and data requirements. For each of the analytical procedures, the pre-outage Morgantown 2 test results was used as an example.
- o Results of the actual steam path inspection and correlation to predicted findings. This enabled identification of problems not properly predicted by the tests and analyses, with subsequent considerations of improved technique (Appendix I).
- o Inspection of steam path components prior to reassembly to determine where hoped-for improvements were not accomplished or only marginally completed (Appendix II).
- o Analysis of post outage test results. This provides an overall calibration of the procedures, identifying the degree of success and pinpointing zones where expected performance improvement did not occur (Appendix III). These would suggest areas for additional work.

PRE-INSPECTION TURBINE PERFORMANCE EVALUATION

The method for performance evaluation is based on the analysis of test information, normal operating records, design information such as heat balances, expected pressures versus flow, section efficiencies and records of previous turbine inspections. Full-scale heat rate tests, while very useful, are not required. The procedure has been categorized into nine basic steps. Each of these will be reviewed in detail and are summarized in Table 7-1.

Pre-Inspection Turbine Performance Evaluation

Section	Suggested Steps Description	Purpose	Information Required
1	Flow Capacity Check (comparison with design)	Determine whether flow capacity is normal, high, low and by how much.	Throttle Flow Measurement; Initial Pressure; Temperature; VWO Heat Balance.
2	Throttle $W/\sqrt{P/v}$ analysis across the load range	Determine changes in nozzle area at tested valve points.	Either heat balances at valve points or baseline test results at valve points; flow measurement, pressure, temperature.
3	VWO flow/1st stage pressure analysis.	Obtain understanding of why flow capacity is what it is. Identify unusual 1st stage pressure problems.	VWO Heat Balance. VWO 1st Stage Pressure. (tested and design). Throttle temperature and pressure. Turbine Cross Section.
4	$W/\sqrt{P/v}$ analyses at HRH, crossover and extraction points.	Determine probable changes in local flow areas due to erosion, deposits, excess leakage or damage. Check probable accuracy of flow measurements. Identify possible turbine bypass conditions.	VWO Heat Balance Test throttle flow and reheat spray flows. Pressure and temperature at each location.
5	Compare test and expected HP section efficiency across the load range.	Determine whether losses are greatest in the 1st stage area or in the following stages. Evaluate the losses due to the following causes: o excessive clearance o deposits o erosion o damage	Enthalpy drop efficiency at valve points across the load range. Heat Balances across the load range. Past Inspection reports. Turbine Cross Section.
6	Compare test and expected IP efficiency levels across the load range.	Determine actual IP efficiency and compare with realistic expected performance. Identify possible bypass or isolation problems. Identify whether excess loss is blamed on LP section.	Heat balances across the load range. Enthalpy drop efficiency across the load range. 1st stage shell pressure comparison with design (only needed if 1st stage shell steam leaks into IP section). Spencer, Cotton, Cannon Paper A Method for Predicting the Performance of Steam Turbine-Generators...16,500 KW and Larger". Measurements of cooling steam flow rates (if cooling steam is employed) Past Inspection Reports.
7	VWO Kilowatt Check	Determine whether the test results and analysis have identified the major sources of loss. Identify possible bypass or isolation problem. Identify whether excess loss blamed on LP section	Measured throttle and RH spray flows; VWO heat balance; HP section losses; IP section losses; probable HP section flow bypass; KW correction for non-standard inlet temperatures, back-pressure and feedwater system operation.
8	Other Considerations Isolation Valve leakage Piston ring or Bell seal leakage	Identify losses and local flow effects to suggest special maintenance procedures.	Heat balances and test measurements on gland steam condenser. Results of procedures 4 and 7, above.
9	Check for probable causes of section losses.	Determine if excessive leakage, deposits, erosion or damage are present. Improve maintenance plan for spare parts and repair procedures.	Enthalpy drop measurements versus time. Diagnostic chart of loss characteristics for different sources. Past inspection reports. 1st stage test pressure versus design across the operating time span. $W/\sqrt{P/v}$ across the time span for all locations. Results of evaluation procedures 1 through 8 (above).

Step 1: Check Unit Flow Capacity

Throughout the evaluation procedures, an accurate value of throttle flow at valves wide open (VWO) conditions will be important. If a VWO throttle flow measurement does not exist, the method presented in Step 4 may be used to estimate the probable flow rate. Where other limitations prevent VWO conditions at rated pressure, tests at reduced pressure with VWO conditions may be attainable. Such tests will be very valuable, for identifying flow capacity, and also for determining HP section enthalpy drop efficiency.

The test flow, W_{test}, should be corrected for deviations from design pressure and temperature as follows:

$$W_{throttle\ corrected} = W_{test} \times \sqrt{\frac{(P/v)_{design}}{(P/v)_{test}}} \quad \text{lb/hr} \tag{7-1}$$

where

v = specific volume

The first use of the corrected W_{thr} is to compare it to design. It is fairly normal for turbines to have flow capacity about 2.5% greater than the VWO heat balance (Note: Be sure to use a VWO heat balance - not a rated heat balance).

Unit Flow Capacity

For Morgantown 2 - Pre-inspection tests (see Figure 7.1).

$$\frac{W_{test\ (corrected)}}{W_{Design}} = 1.135, \text{ or } 13.5\% \text{ excess flow}$$

Since turbine manufacturers typically exceed the design flow values by about 2.5%, this represents about 10.7% greater flow capacity than normal. Values outside the 2.5% greater-than-design flow rate should raise several questions:

If test flow is high -

- o can the turbine first stage nozzle be eroded? (known to be a highly likely condition for this unit)

- can 1st stage shell pressure be low due to excessive packing clearance or enlarged 2nd stage area?
- can leakages and bypasses exist in piston rings, bell seals, start-up drain valves, etc.?
- can HP feed valves to BFP turbines and steam seal systems be open when they are supposed to be closed?
- can the flow measurement be wrong?

If the test flow is low -

- can the 1st stage nozzle be battered closed?
- can the 1st stage pressure be high?
- can the flow measurement be wrong?

The above questions will be addressed as the subsequent sections are completed.

Step 2: Throttle $W/\sqrt{P/v}$ Across the Load Range

Several flow relationships will provide immediate information regarding the internal conditions of the turbine. The basic flow Eq. 7-2 is correct for circumstances where local stage pressure ratios are constant. If erosion, damage or other problems cause changes to local areas, the equation is no longer accurate. However, it can be used to recognize that a change has occurred and to help estimate the magnitude of change.

$$W = K \quad A \quad \sqrt{P/v} \tag{7-2}$$

where W = the flow to the following stage

$K\,A$ = a value proportional to effective flow area

P = throttle pressure (absolute)

v = throttle specific volume

The equation can be rearranged to solve for A.

$$A = \frac{W}{K\sqrt{P/v}} \tag{7-3}$$

In this way, where W (flow to the following stage), P and t are known, it is very easy to determine whether changes in Area have occurred (for most circumstances the proportionality constant, K, can be ignored).

The purpose of this analysis is to determine the changes in 1st stage nozzle area at the various valve arcs. The desirable starting point is a plot of design flow values (at valve points) versus $W/\sqrt{P/v}$. Note that since P and v ahead of the throttle valves are normally constant, this will simply be a straight line plot as shown in Figure 7-1.

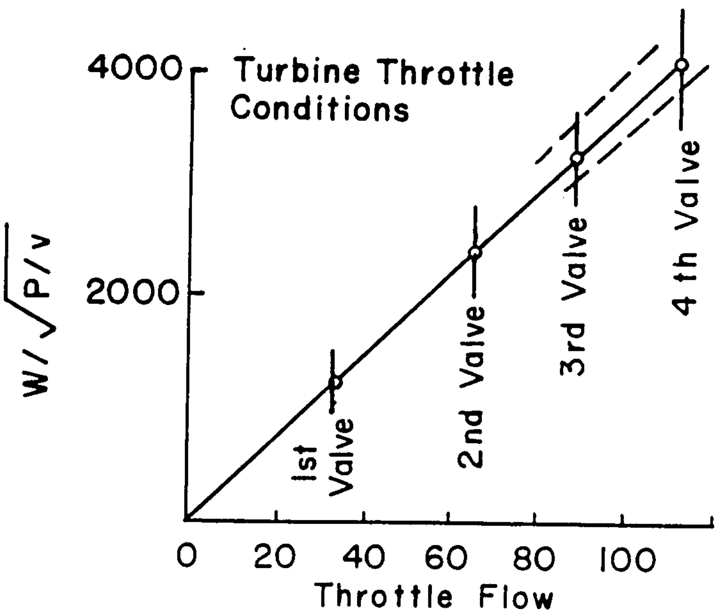

Figure 7-1

Unfortunately, many of the heat balances are made for convenient percentages of throttle flow (i.e., for 40, 60, 80 and 100% flow) and not at discrete valve points. In addition, experience has shown that MHC turbines commonly have their control valves set to improperly follow the design throttle flow versus valve position criteria. This can have significant impact on response, high pressure turbine efficiency and diagnosis of early valve arc flow areas.

When valve point heat balance data is unavailable, baseline test data obtained at the initial start-up or after turbine overhaul may be substituted.

Figure 7-2 illustrates actual test data (Morgantown 2) plotted against the design data at valve points. The corrected test $W_{throttle}$ with design pressure and specific volume should be used in this plot (see Eq. 7-1).

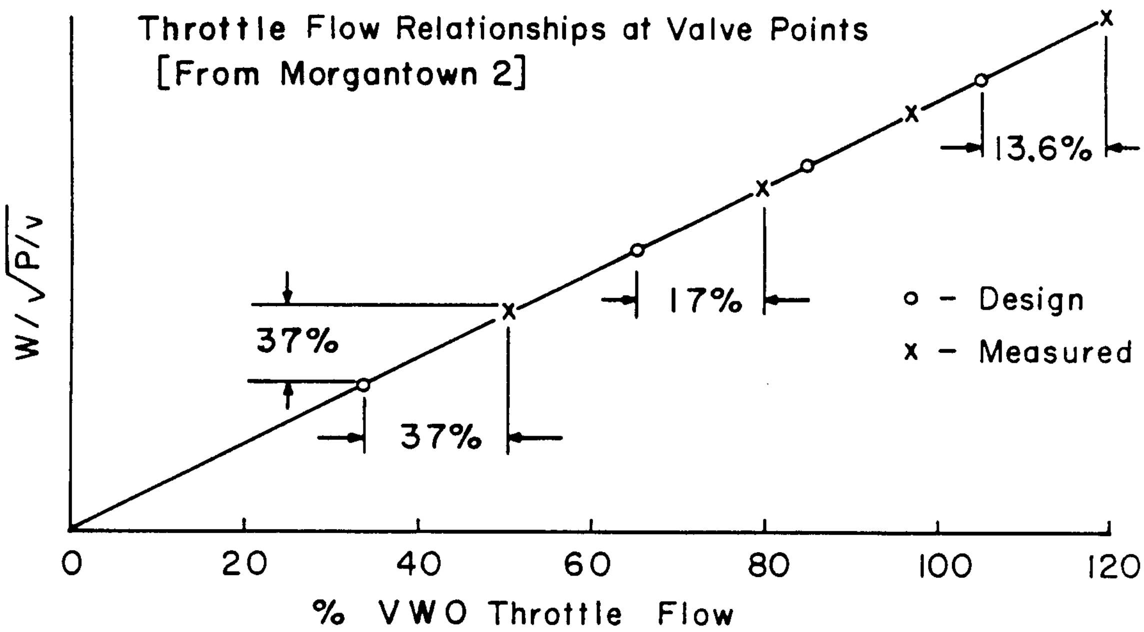

Figure 7-2

It is apparent that the first valve arc flow has increased by about 37%. At full load, the flow increase over design appears to be about 13%.

It should be recognized that the stage flow passing ability is affected not only by area but also by the stage pressure ratio. The stage flow is also affected by off-angle flow conditions, changed reaction, damage to profiles and other physical changes to the steam path. Most stages of the turbine operate with constant pressure ratios across the load range. But the first stage has a highly varying pressure ratio on those turbines utilizing partial arc operation.

As long as the 1st stage nozzle pressure ratio approaches or is greater than acoustic (1.83) the throttle flow will be directly proportional to area. The nozzle pressure ratio at valve points can be crudely determined as follows:

P_{bowl} (pressure ahead of nozzle) $= P_{throttle} \times 0.96$ (7-4)

P_2 (pressure downstream of nozzle) $= P_{1st\ stage\ shell}$ (7-5)

(Note: This assumes that the 1st stage reaction is zero. This is an inaccurate assumption but will not significantly compromise the calculation being made here.)

These relationships are roughly sketched in Figure 7-3 for a six-valve unit.

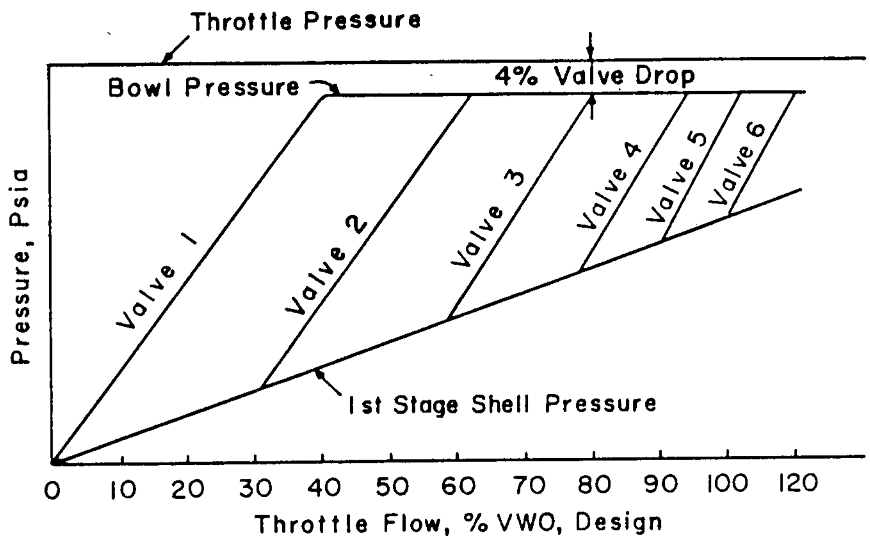

Figure 7-3

Where test flow is different than design (or baseline) flow, the approximate change in nozzle area for each valve point can be determined as follows:

$$\frac{A_{test}}{A_{design}} = \frac{W_{test(corrected)}}{W_{design}} \times \frac{N_{design}}{N_{test}} \quad (7\text{-}6)$$

where N is the restriction factor, a function of the nozzle pressure ratio, as shown in Figure 7-4.

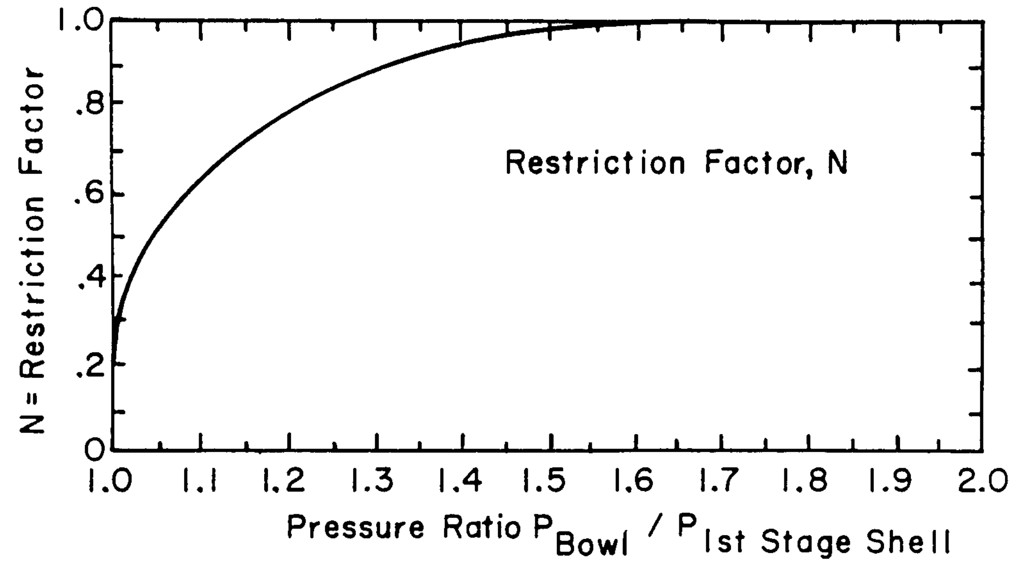

Figure 7-4

Note that when sonic conditions exist (N=1), area is directly proportional to flow.

In the case of Morgantown 2, the first and second valve points do have critical nozzle pressure ratios and the change in area can be quickly determined from the increase in measured flow over design flow:

$A_{1st\ valve}$ = 1.37 x design

$A_{1st\ +\ 2nd\ valve}$ = 1.17 x design

$A_{4\ valves\ open}$ - see Step 3 below

Keeping in mind the possibilities of piston ring leakage or other bypassing flow conditions, it would appear that the first valve nozzle has increased 37% in area and the second arc is unchanged. (The 37% change in the lst arc alone is enough to explain a 17% increase when both 1 and 2 are open.) It is likely that the third and fourth arcs are unchanged also, but this calculation should be held until the first stage shell pressure effects are discussed (next section).

Step 3: VWO Flow Versus 1st Stage Shell Pressure Analysis

Figure 7-5 shows a comparison of design versus measured pressure for various throttle flow conditions as observed at Morgantown 2. Pressure is plotted instead of $W/\sqrt{P/v}$ since specific volume (v) can rarely be determined for this turbine location.

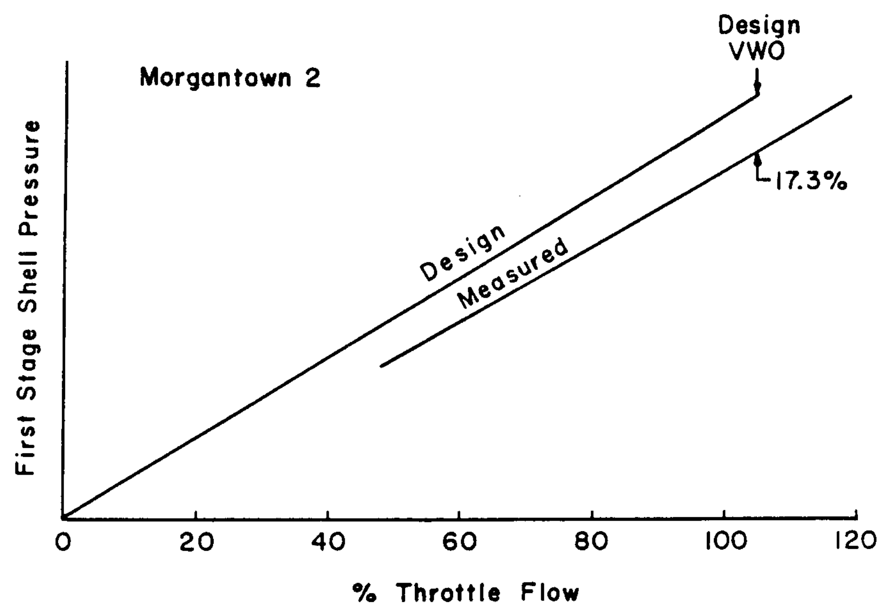

Figure 7-5

If 1st stage shell pressure is low for a given level of throttle flow, it can mean one or more of several things:

- o adjacent packings (such as N2, 2nd stage diaphragm or HP dummy) may be badly rubbed
- o 2nd stage nozzle areas may be enlarged
- o measured flow may be exaggerated
- o other leakages or bypasses may exist

If the pressure is high:

- o 2nd stage nozzle areas may be diminished
- o throttle flow measurement may be underestimated

It can be seen that for Morgantown, the measured first stage shell pressure is approximately 17% below design at any given throttle flow. This lower pressure cannot be explained by an increase in second stage area or by packing rubs alone, since neither can cause that great an effect. These data suggest that both are enlarged and, further, that the main steam inlet piston ring seals may be leaking as well.

Using Eq. 7-6:

Design Conditions:

$P_{1st} = 2660; \; P_{Bowl} = 3374$ $\qquad \dfrac{P_{Bowl}}{P_{1st}} = 1.268$

N = 0.85 (from Figure 7-4)

Test Conditions:

$P_{1st} = 2550; \; P_{Bowl} = 3374$ $\qquad \dfrac{P_{Bowl}}{P_{1st}} = 1.323$

N = 0.89

$$\frac{A_{test}}{A_{Design}} = \frac{W_{test}}{W_{Design}} \times \frac{N_{Design}}{N_{Test}}$$

$$= 1.136 \times \frac{0.851}{0.890} = 1.086$$

This would appear to be a good check that only the first valve arc of nozzles is eroded. The total area appears to have enlarged about one-fourth the increase of the first valve arc.

Determination of which conditions exist must be deduced from this and subsequent sections. Keep in mind that past internal problems often reoccur. Past inspection reports that indicate heavy packing damage, for example, would increase the odds that rubbed packings have again contributed to low 1st stage pressure.

Note that low 1st stage pressure, in addition to increasing turbine flow capacity will increase the available energy on the 1st stage. This lowers its velocity ratio and efficiency while taking energy away from the more efficient later stages.

Step 4: $W/\sqrt{P/v}$ Analyses at Hot Reheat (HRH), Crossover and All Extraction Points

As indicated previously the relationship $W/\sqrt{P/v} = K\,A$ is essentially a calculation for area. It can be used in three basic ways:

1. to recognize that flow areas have changed;
2. to deduce approximate local flow rates (only where flow areas are known to be unchanged).
3. to validate test data (i.e., plot should be flat over the entire load range).

At the hot reheat point, both erosion and damage are frequently present. At the cold reheat point, pressure is usually lower than the design heat balance values due to lower actual reheater pressure drop, also due to 1st reheat stage erosion. At locations below this point (extractions and crossover), design flow conditions usually (but not always) exist.

One common complication is the physical location of test measurements for P and t. These frequently provide different conditions from those used on the design heat balance. For example, pressure is usually measured in the extraction pipe a distance from the turbine instead of immediately after the stage as shown on the heat balance. A correction of about +3% is normally sufficient to correct for this, however if the measurement is made further downstream such as at the heater, a +6% correction may be required.

The temperature (enthalpy) at extraction points is usually hotter than that in the steam path by about 15 BTU. This occurs due to the temperature segregation (from spillstrip leakage) that causes a ring of hot steam to exist where the extraction steam is drawn off (this would not occur for extractions at the cold reheat or crossover points). The magnitude of high enthalpy in the extractions should be checked by drawing a section expansion line from the bowl to the exhaust, then spotting the extraction enthalpy (corrected back to the pressure at the stage). The specific volume used in the relationship $W/\sqrt{P/V}$ should be determined from the conditions at the steam path, thus better representing the steam that passed through the following stages instead of the steam drawn out the extraction point (about 15 Btu colder than the extraction). By doing this, an improved $\sqrt{P/v}$ can be obtained for use in the test $W/\sqrt{P/v}$ calculation for comparison with design. Note that if the test extraction is significantly hotter than the 15 BTU mentioned above, then leakage into the extraction pocket should be suspected.

Another complication is the desirability to know the flow to the following stage. A preferred basis for $W/\sqrt{P/v}$ is where W is flow to the following stage. Where this is not known and where the feedwater system is known to be operating normally, it is usually satisfactory to use throttle flow at all points for both design and test. For points at or below the reheat point, correction should be made for reheat attemperation. If the test spray flow is 3%, for example, the value of W_{thr} (throttle flow) should be increased by 3% for calculations at the HRH point and below.

As an example of using $W/\sqrt{P/v}$ for determining probable throttle flows, the following unit had zero spray flow and was known to be free of erosion or damage in the IP section (not Morgantown 2).

Table 8-2

Location	Test $\sqrt{P/v}$	Design $W_{thr}/\sqrt{P/v}$ ($\times 10^{-3}$)	Deduced Throttle Flow = (Test $\sqrt{P/v}$) ($W_{thr}/\sqrt{P/v}_{Des}$)
HRH	19.434	212.2	4.1238×10^6 lb/hr
9th Stage	10.59	392.5	4.158×10^6 lb/hr
LP Bowl	6.68	606	4.047×10^6 lb/hr
AVERAGE			4.1097×10^6 lb/hr

The average was used in subsequent calculations for comparing design and test flow; design and test output; and design and test first stage pressure.

Use of $W/\sqrt{P/v}$ for Determining Changes in Flow Area:

Where flow area has increased, for instance due to erosion, the value $W/\sqrt{P/v}$ will increase. The relationship is far from linear, however, since the percentage change in area will be far greater than the percentage change in $W/\sqrt{P/v}$.

Figure 7-6, below, can be used to interpret changes in flow area based on $W/\sqrt{P/v}$ effects.

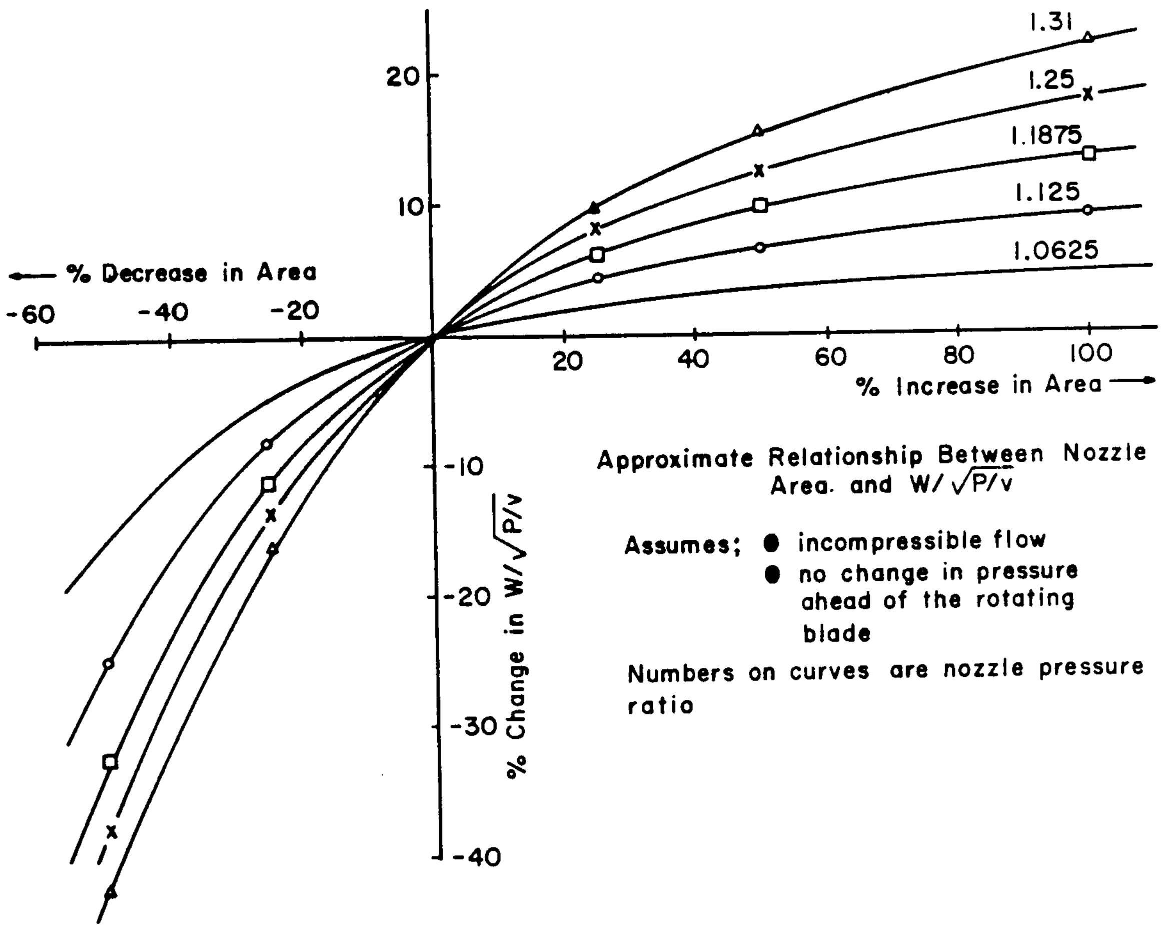

Figure 7-6

This figure assumes that the area change is confined to one stage; that the rotating blades are not affected; and, further, that the pressure ahead of the rotating blades has not changed. These assumptions are not precise, but usually can give reasonable results for estimating area changes prior to opening the unit. Note that if there is a flow exaggeration or a flow bypass, $W/\sqrt{P/v}$ will appear large and the deduced area enlargement will also be exaggerated.

Several additional suggestions are listed below:

1. The lowest two extractions usually show serious deviation from design and should not be used to determine flow. They also should not be trusted to predict area.

2. Patterns of test $W/\sqrt{P/v}$ deviation from design may have significance. For example, if the BFP turbine is extracting excessive flow from the main steam pipe at VWO flow conditions and then discharging portions of it into the crossover area, the HRH and top IP section $W/\sqrt{P/v}$ may appear high, while the crossover may appear normal.

3. A general pattern of high $W/\sqrt{P/v}$ values may either indicate the throttle flow is exaggerated or that steam is bypassing the turbine. Checking KW output and first stage shell pressure may provide additional clues.

4. If a heater is out of service, or performing poorly, the next higher heater pressure will be low and $W/\sqrt{P/v}$ values will appear high.

In the case of Morgantown 2, the pre-outage tests indicate the following:

Location	Design $W/\sqrt{P/V}$	Test $W/\sqrt{P/V}$	$\frac{\text{Test-Design}}{\text{Design}}$
HRH	162	193	0.19
#2 Extraction	345	379	0.10
Crossover	492	590	0.20

The above suggests that the first reheat and first LP stages are greatly eroded open and that even the middle IP stages are significantly enlarged.

Serious doubts were felt about these numbers:

- The crossover pressure measurement was suspect.
- Flow bypassing of the turbine was suspected.
- Exaggeration of the flow by the flow nozzle was suspected.
- Increased demand for flow by the boiler feed pump turbine was believed likely.

In spite of the above concerns, preparations for repairs of seriously eroded nozzles was recommended.

In regard to the flow nozzle, it is generally common in existing plants to find the flow measurement with a variety of faults:

- No flow straightener
- No calibration

- Close-up turns or pipe fittings
- Not inspectable

The usual result is an exaggeration of the actual flow — even 3-5% can be expected. This tends to make heat rate appear poorer than actual and also complicates the diagnoses of internal problems. Where maximum plant loading is established by the flow measurement, it also limits the unit output to a smaller value than was intended. At Morgantown 2, a new flow section was to be installed during the inspection to eliminate the above concerns.

Step 5: Compare Test Versus Expected HP Section Enthalpy Drop Efficiency Across the Load Range

The most important single test point for evaluation is VWO, even where throttle pressure has to be reduced to enable this test. Comparison with VWO heat balance values can be made directly, but one should be aware that manufacturers expect some margin of HP efficiency with the heat balance values — typically 2.5% when the unit is in near-perfect condition.

Where lighter load tests are available, even more can be deduced. The performance (both design and test) is easier to diagnose if plotted against percent valve position, but is also effective if plotted versus throttle flow. Plotting against $P_{CRH}/P_{throttle}$ or $P_{1st}/P_{throttle}$ often makes diagnosis difficult because both P_{CRH} and P_{1st} deviate from design due to reheat nozzle changes, low reheater ΔP, and excessive leakage from the 1st stage shell area.

Figure 7-7 shows a hypothetical plot of HP section efficiency versus control valve position (only the valve point data is shown).

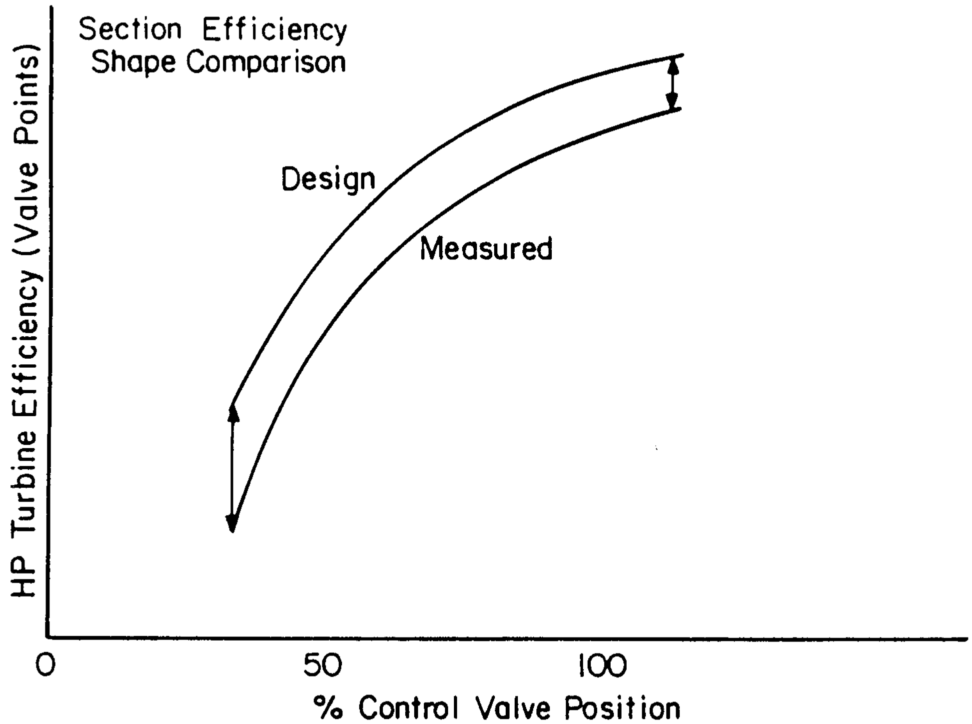

Figure 7-7

Note that the difference between design and test is greater at light load than at VWO. This is generally a sign that the first stage performance has deteriorated more than the later stages. This would not be true, of course, for a unit with throttling valve control or full-arc admission.

Where the opposite occurs (less deterioration at light load), it usually indicates the latter stages are more affected than the first stage.

The turbine cross section should be carefully examined to see if significant leakages are directed to the HP exhaust. For example, separate single flow HP sections usually have a major packing that allows leakage steam to mix with the cold reheat flow. Poor section efficiency and low 1st stage shell pressure may be the result of excessive leakage through that packing.

Another example would be leaky main steam piston rings (or bell seals). The turbine cross section will indicate whether such leakage will go to the cold reheat, hot reheat, or crossover areas. Such leakages (at throttle enthalpy) will vary across the load range depending on which rings leak and what valves are open. This can have shape effects on HP section efficiency. Note the actual example in Figure 7-8 which is believed caused by such leakage.

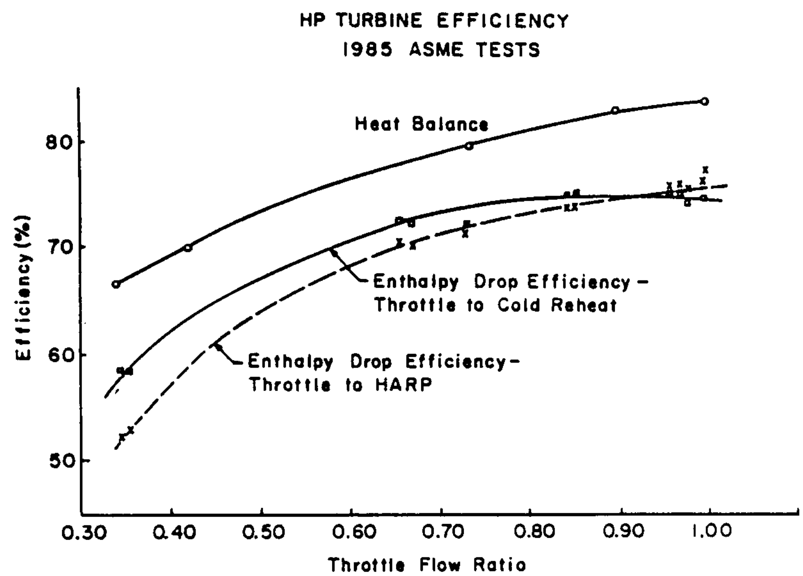

Figure 7-8

The dotted curve, enthalpy drop efficiency from throttle to the HARP (heater above the reheat point), shows the most proper shape. It would be unaffected by piston ring leakage that has probably affected the measured entire section enthalpy drop efficiency. In addition, since its end point is based on extraction enthalpy, the deduced efficiency should be lower than that of the entire HP section. Since it is higher (at VWO conditions), one should further suspect that hot leakage steam into the cold reheat area is affecting both the level and shape of the measured overall section efficiency.

The HP section kw losses should be determined at VWO as illustrated in the following equations:

$$\text{Kw Output}_{\text{Design}} = \frac{(H_{\text{throttle}} - H_{\text{CRH}}) \cdot W_{\text{throttle}}}{3412} \tag{7-7}$$

$$\text{Kw Loss} = \frac{\text{\% HP Loss x Kw Output Design}}{100} \tag{7-8}$$

Be sure to correct HP losses to a percentage:

$$\text{\% HP Loss} = \frac{\text{Efficiency}_{\text{Des.}} - \text{Efficiency}_{\text{Test}}}{\text{Efficiency}_{\text{Des.}}} \tag{7-9}$$

Small flow corrections can be made for flows that pass through the 1st stage then leak into the reheat section; also for HARP flow that does not pass through the last few HP stages. Note that the Kw losses calculated here do not include the effect of excess throttle flow. A further correction will be made later for the Kw check analysis.

In general the HP section performance will deteriorate much more than the other sections. Usually the greatest source of such loss will be rubbed seals and excessive leakage, with solid particle erosion the second largest loss.

In the case of Morgantown 2, Figure 7-9 below shows tested, heat balance and expected (heat balance plus margin) high pressure performance.

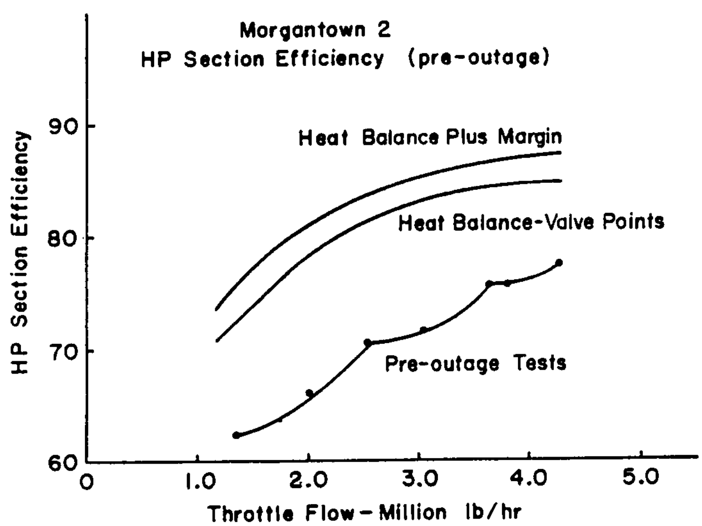

Figure 7-9

At WO, the test efficiency is 7.5 points poorer than the heat balance and 10 points poorer than expected. At the 1st valve point, the tested efficiency is about 10 points poorer than heat balance and 12.5 points poorer than expected.

This suggests that more degradation is present at the first stage than the latter HP stages. This is consistent with the expected heavy erosion.

The approximate expected HP output in kilowatts is:

$$KW = \frac{\text{Des. Throttle Flow } (H_{thr} - H_{crh})}{3412}$$

$$= \frac{3975497\ (1424 - 1266.5)}{3412} = 183,511$$

HP loss (from new condition):

$$\%\ \text{Loss (at des. flow)} = \frac{\text{Efficiency}_{exp} - \text{Efficiency}_{test}}{\text{Efficiency}_{exp}}$$

$$= \frac{0.875 - 0.775}{0.875} \times 183,511 = 20973\ \text{Kw}$$

But note that the loss from heat balance conditions for output comparison with tests is less by the amount of HP section margin.

$$\frac{0.85 - 0.775}{0.85} \times 183,511 = 16192\ \text{Kw}$$

Also, note that at the higher than heat balance test flow, the HP Kw losses will increase to:

$$\frac{W_{test}}{W_{Des}} \times 16192 = 1.0704 \times 16192 = 17331\ \text{Kw}$$

Step 6: Compare Test Versus Expected IP Section Enthalpy Drop Efficiency

A series of complications make diagnosis of most IP sections difficult. These include:

1. Heat balances usually suggest poorer performance than is really expected. Figure 7-10, from GE's paper "A Method for Predicting the Performance of Steam Turbine-Generators...16,500 KW and Larger" can usually be helpful in determining what is a reasonable expected level for units built since 1960. Note that Figure 7-10 is for performance from the IP bowl to the IP exhaust. Subsequent

corrections for valve and crossover pressure drops are needed to make direct comparison with test results.

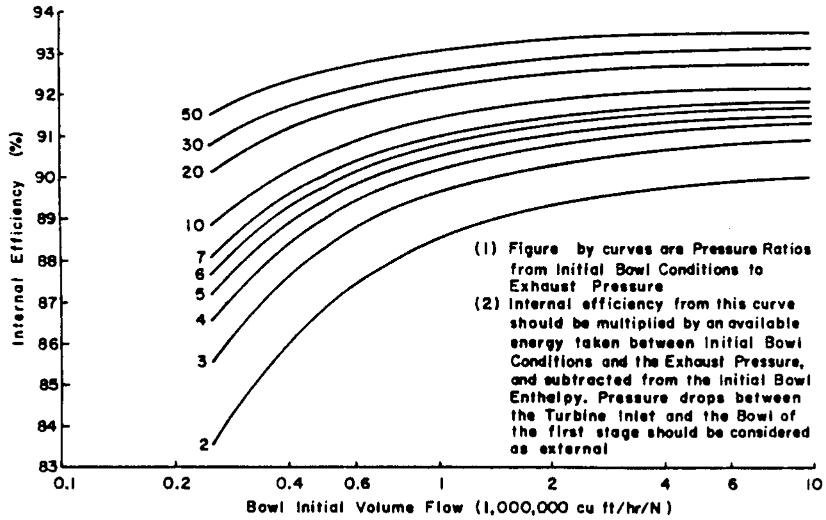

Figure 7-10

2. N2 or IP dummy packing flows on opposed flow turbines (combined HP and IP sections on a single shaft span) will cool the IP bowl. These low enthalpy flows decrease the IP bowl enthalpy and cause an apparent increase in IP section enthalpy drop efficiency. To the best degree possible, this effect must be eliminated so that the true IP efficiency can be determined. These methods will be discussed later.

3. Many double flow IP sections have shaft cooling steam flows. These, too, cause the IP enthalpy drop efficiency to appear higher than its actual performance level. Where practical, the cooling flow rate and enthalpy should be measured or estimated to help determine the IP bowl enthalpy.

4. The crossover area may receive a variety of other flows, including:

   o excess flow from BFP turbine exhaust

   o leakage from dummy packing

   o leakage from bell seals, piston ring seals

   As above, these flows, with their enthalpy, need to be measured or estimated.

The most critical of the above complications is the N2 packing on GE opposed Flow units and the IP dummy flow on W units. The N2 arrangement is somewhat easier to diagnose from test results.

Excess N2 packing leakage will cause an apparent increase in IP section enthalpy drop efficiency as shown in Figure 7-11 below. This effect is even more noticeable at light load. Since real IP section efficiency is virtually constant across the load range, it is often possible, by trial and error, to determine a magnitude of packing clearance that would explain the deviations of IP enthalpy drop efficiency from the expected flat condition.

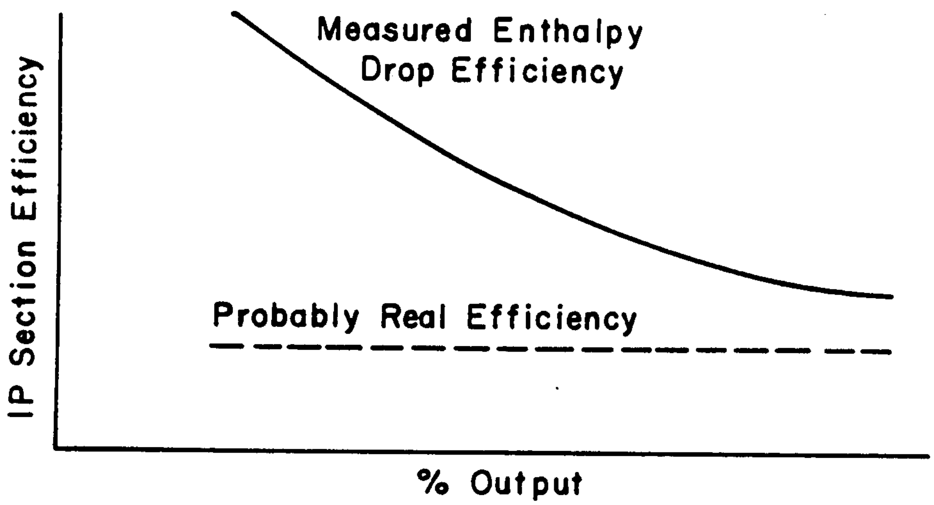

Figure 7-11

The trials are based on multiples of heat balance N2 flow, with success being achieved when the corrected efficiency is constant across the load range. The method is marginally successful at best. One problem area is the N2 enthalpy, which may not be as shown on the heat balances and will deviate differently at high and low load conditions.

The GE paper "Estimating the Leakage from HP to IP Turbine Sections" by John A. Booth and David E. Kautzmann, Appendix IV suggests three methods for estimating flows from HP to IP sections. These are well presented and worth consideration with the following comments:

- o The "Blowdown System" (System A) is not recommended due to its several disadvantages:
  - Creates excessive stress on packing teeth.
  - Dumps more steam to the condenser than the desuperheater may be designed to handle.
  - Indicates an enthalpy of the flow that would not be identical with that of normal N2 flow.

- Figure 7-12 is taken from the Booth and Kautzmann paper, with P_a (blowdown annulus pressure) added for clarity.

  When large, high velocity flows are being passed through the blowdown valve, the pressure drop from P_a to P_3 can be large. This is a zone with fairly small flow areas and tight turns. It can be understood, then, that P_a will be higher than P_3. The assumed condition of Q_2 (flow from the IP bowl to the blowdown annulus) may not exist even though P_3 is lower than P_2.

o The "more elaborate" system acknowledged in the same section is being tested at Morgantown 2 and a report will be issued later on its results including a comparison with system (B).

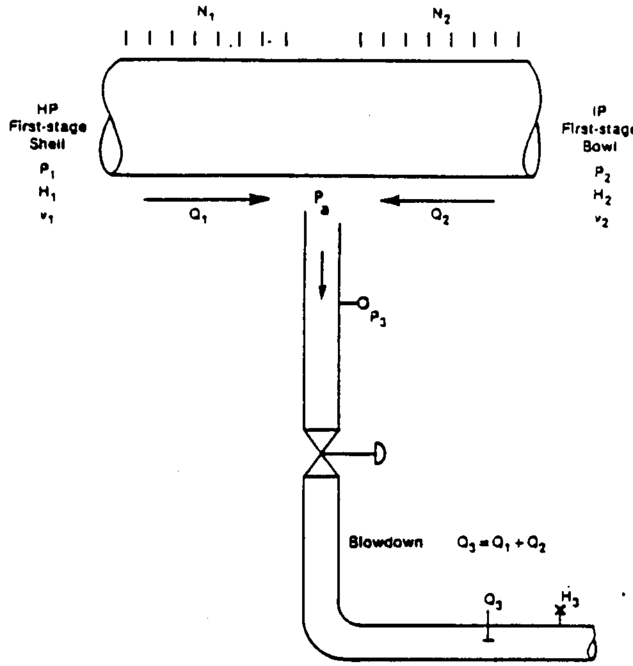

Figure 7-12. **Blowdown system**

Morgantown 2 IP Section Tests

In the case of Morgantown 2, the measured (apparent) efficiency (intercept valve to crossover) is shown below on Figure 7-13. The design values are based on Figure 13 of the GE paper, but corrected to intercept valve pressure.

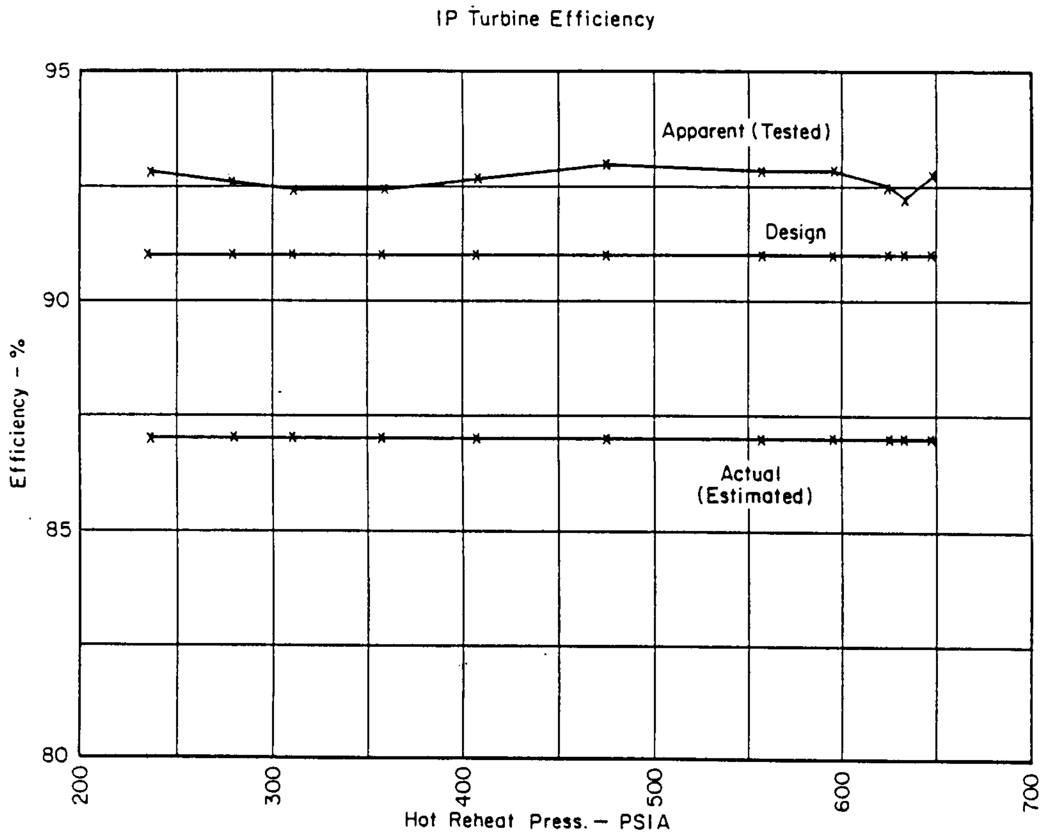

Figure 7-13

It was estimated that with the suspected level of nozzle erosion and seal wear, that the real IP efficiency would be no better than 87%.

Based on the large difference between test efficiency and assumed actual efficiency, estimates were made for the possible combination of excessive N2 leakage and mainsteam piston ring leakage that could cause such a great impact.

Leakage rates up to 500,000 lb/hr were considered. Such flows could explain both the low 1st stage shell pressure and the high IP section efficiency, but were felt to be unreasonably great. But recommendations were made to be prepared for (and inspect) all main steam piston rings, to replace all N2 packing,and to add an N2 blowdown bypass system capable of measuring the effective N2 clearance during operation.

Like the HP section, the design IP section output can be determined from the heat balance:

$$Kw_{output} = \frac{\text{Reheat Flow } (H_{iv} - H_{xo})}{3412}$$

where flow can be roughly established by averaging the flow entering the IP section and the flow leaving.

In the case of Morgantown 2:

IP section Kw output = 161850 Kw

The Kw losses in the IP section at VWO flow are:

$$\frac{\text{Efficiency}_{\text{Expected}} - \text{Efficiency}_{\text{Test}}}{\text{Efficiency}_{\text{Expected}}} \times \text{design output}$$

$$\frac{0.04}{0.91} \times 161850 = 7100 \text{ Kw}$$

(At test flow, the loss is increased to 7500 Kw.)

Step 7: VWO Kilowatt Check

Knowing test flow, test losses in the HP and IP sections and test kilowatts, it is appropriate to compare output to heat balance values. One should recognize that there are a lot of unknowns in this process. LP section performance is unknown. N2 or dummy flows at best have been estimated. Testing inaccuracies may be great. Still, the check can be extremely valuable. Note that proprietary analytical programs such as PEPSI can be used in similar fashion.

It is desirable to correct the test output for unusual back pressure, reheat spray, reheater ΔP, serious feedwater heater deviation from design conditions and reheat temperature. It is usually best not to make corrections for initial pressure and temperature.

If throttle flow is 5% higher than the heat balance, one would naturally expect about 5% greater output. However keep in mind that greater flow usually increases CRH pressure, thus decreasing the HP available energy (AE) and used energy. For large flow deviations it is desirable to correct for this effect using test CRH pressure to compare AE_{test} with AE_{Des}.

Often the CRH pressure will be low, due to smaller-than-design reheater ΔP and HRH nozzle erosion. This will decrease the correction for reduced HP section AE caused by excessive flow. Note that the combined effect of excess flow and low reheater ΔP can be simultaneously considered by comparing test CRH pressure and AE with the heat balance. The thermal kit correction for RHTR ΔP would tnen be unnecessary.

An example of a simple Kw check procedure (not Morgantown 2) is shown below:

	% effect	Effect on Output (MW)
HP Turbine		
Throttle flow: down from HB by 4.7%		-5.0
Low HP efficiency and excessive N2 Leakage		-15.1
IP Turbine		
IP flow (including sprays): up by 4.5%		+12.5
Low IP efficiency		-4.2
Other KW Effects		
Low IP bowl enthalpy (mixing)	-0.25	-1.0
Low HRH Temp.	-0.25	-1.0
High B.P.	-4.75	-19.3
BFPT Excess Flow	-1.4	- 5.7
Total change in output		-38.8
Compare with heat balance and test		
Heat Balance - Test Output		-42.1

The above suggests that the losses have been approximately pinpointed by the tests and analyses. Some unknown or LP losses can be expected. However, emphasis should be placed on the necessity to confirm that the local losses are as predicted when the turbine is open. This is to provide assurance that no significant cause of loss is overlooked.

Note that if large unexplained losses had occurred, suspicion would have been placed in a number of areas:

- o Incorrect flow measurement -

  Consider questions raised in previous section regarding $w/\sqrt{P/v}$ analyses

- o Poor LP section performance -

  Do past reports indicate deposits, damage or serious rubbing has been previously observed? Can deposits be checked on the L-0 nozzle and bucket during a weekend shutdown? Is there reason to suspect that the LP turbine is really poorer than is expected by the heat balance?

  LP turbines usually are relatively free from losses caused by solid particle erosion and foreign material damage. Even excessive rubbing does not normally cause serious losses. Whenever an analysis indicates the probability of more than 2% LP losses, suspicion should be raised that something else may really be wrong.

- o Serious N2 or IP dummy packing leakages -

  Does the IP section efficiency seem artificially high and/or increase as load is decreased? Do past reports show heavy rubs to be a problem?

- o Unit is not isolated - flow is bypassing.

<u>Kilowatt Check - Pre-outage - Morgantown 2</u>

5% Overpressure VWO Heat Balance Kw	625,496
Measured main steam test flow is higher than the VWO 5% OP HB by 2.2%	
Increased HP output due to 2.2% extra flow =	+ 4,070
Increased HP output due to low CRH pressure (combined effect of low RHTR ΔP and eroded HRH nozzle)	+13,007
HRH flow is higher than HB by 7.8%	
Increased RH section output =	+34,360
Loss due to 8.8% poor HP section	-17,331
Loss due to excess N2 flow (100,000 lb/hr)	- 3,957
Loss due to poor IP section (down 4%)	- 7,654
Other cycle corrections considered negligible	

Expected Test Output	647,991
Actual Test Output	621,000
Unexplained Loss	26,991 (4.3%)

This unexplained loss can be assigned to a combination of:

- o test flow exaggeration
- o Cycle flow bypasses (turbine flow bypass, piston ring leakage, etc.)
- o Deteriorated LP section efficiency
- o Greater N2 flow than assumed

It is becoming apparent that there must be a combination of flow error and a cycle bypass affecting the diagnosis and output of this unit. These will be discussed later.

Step 8: Other Losses

Experience has shown that a large variety of leakages can impact strongly on the real output and efficiency of the unit. Most of the time these leakages do not reveal themselves in poor section efficiency. Where such leakages exist, the analytical methods described above would generally indicate that there must be flow errors or that the LP section must have poor efficiency.

An excellent reference on this subject "The Best Buy in Heat Rate Recovery - Turbine Cycle Isolation Maintenance" by W.H. Hopson, J.C. Peyton and J.K. Legg of Southern Company Services is included in Appendix IV.

Morgantown 2

Examination of the previous sections shows a consistent pattern:

- o 1st stage pressure is unusually low for the measured throttle flow. Explaining this lowness by normal methods (high N2 flow; high 2nd stage diaphragm packing flow) requires such extreme clearances that they seem unbelievable.
- o $W/\sqrt{P/v}$ analysis showed consistently high values that suggest the deduced flow must be exaggerated.
- o The kilowatt check shows unexplained losses that suggest the deduced section flows must be exaggerated.

While no measurements existed to confirm the magnitude of such flows, it was concluded that two bypass conditions existed that had reasonable probability:

- A start-up bypass around the turbine was known to be hot when it should have been cold. A level of 100,000 lb/hr was assumed (2.4%).
- The mainsteam turbine inlet piston rings were believed the most likely candidate for a second leak. 117,000 lb/hr was assumed to bypass the high pressure stages and re-enter at the cold reheat point.

These two potential leaks, if true, would cause over 21,000 Kw of the 27,000 Kw unexplained loss.

Step 9: Determine the Probable Causes of Section Losses

The previous sections generally identify efficiency levels, magnitudes of loss, probable changes in local flow areas. The next section deals with the probable causes of such effects.

To use the techniques described below, it is desirable to plot parameters that deviate significantly from design as a function of time. This will allow determination of whether changes have happened suddenly or slowly.

Diagnosis of Problem Areas

Four major areas of common trouble can be expected to afflict large steam turbines including:

- Excessive Leakage
- Erosion (solid particle)
- Internal Damage
- Deposits

Each of these potential troubles has characteristics that will enable the diagnostician to differentiate using the available monitoring results.

Differentiating Characteristics:

Table 7-3 is a summary of the symptoms normally present for each of the common problems. This brief listing will help diagnose turbine internal conditions.

Table 7-3

DIAGNOSTIC CHART OF LOSS CHARACTERISTICS

Rubbing Damage on Spillstrips and Packing	
Mode of Appearance:	Happens suddenly - more likely on a first startup.
Local Effects:	Increases flow capacity (this effect highest in HP section). Decreases section efficiency (worst on low volume flow stages). May cause IP enthalpy drop efficiency to appear higher (opposed-flow units only).
Side Effects:	Worsens flow temperature segregation. Normally has little effect on thrust.
Shape Effects:	Ratio of % ΔEfficiency/% ΔFlow usually greater than 1. (Absolute values)
Special Dangers:	

Solid Particle Erosion	
Mode of Appearance:	Usually appears gradually.
Local Effects:	Increases flow capacity. Decreases efficiency. Worst effects usually at turbine inlets; at first stage, erosion magnitude may be worst at the inlet fed by the first valve.
Side Effects:	Changed thrust; changed $\sqrt{P/v}$ distribution; changed flow distribution.
Shape Effects:	$\sqrt{P/v}$ effects may be greatest at light load. Efficiency loss compared to guarantee may be greatest at light load; thrust increase may be in the same direction as flow.
Special Dangers:	Overloaded buckets; weakened tenons.

Table 7-3 Continued

DIAGNOSTIC CHART OF LOSS CHARACTERISTICS

Deposits	
Mode of Appearance:	Usually gradual; may reach a self-limiting magnitude, then not increase further; may appear to decrease following a shutdown or major temperature swing.
Local Effects:	Decreased efficiency; decreased flow capacity.
Side Effects:	Changed thrust; changed $\sqrt{P/v}$ distribution.
Shape Effects:	Section efficiency may decrease 3-4 times as much as flow capacity. Thrust changes may be opposite the direction of flow.
Special Dangers:	Excessive thrust.

Internal Damage	
Mode of Appearance:	Usually abrupt - may have subsequent symptoms.
Local Effects:	Decreased efficiency; decreased flow capacity.
Side Effects:	Increased vibration; changed $\sqrt{P/v}$ distributions; changed thrust.
Shape Effects:	No consistent pattern.
Special Dangers:	Weakened or loosened mechanical structures.

The losses from the above causes are highly variable. A range of typical conditions found on inspection are shown below:

Typical Loss Magnitudes

	Solid Particle Erosion	Deposits	Internal Damage	Rubbed Seals
HP	0 - 2%	0 - 10 %	0 - 3%	2 - 12%
IP	0 - 2%	0 - 5%	0 - 2%	1 - 4%
LP	0. - 0.5%	0 - 3%	0 - 1%	0 - 1%

Examples of Problem Diagnosis

Excessive Leakage Caused by Shaft Rubbing (see Figure 7-14)

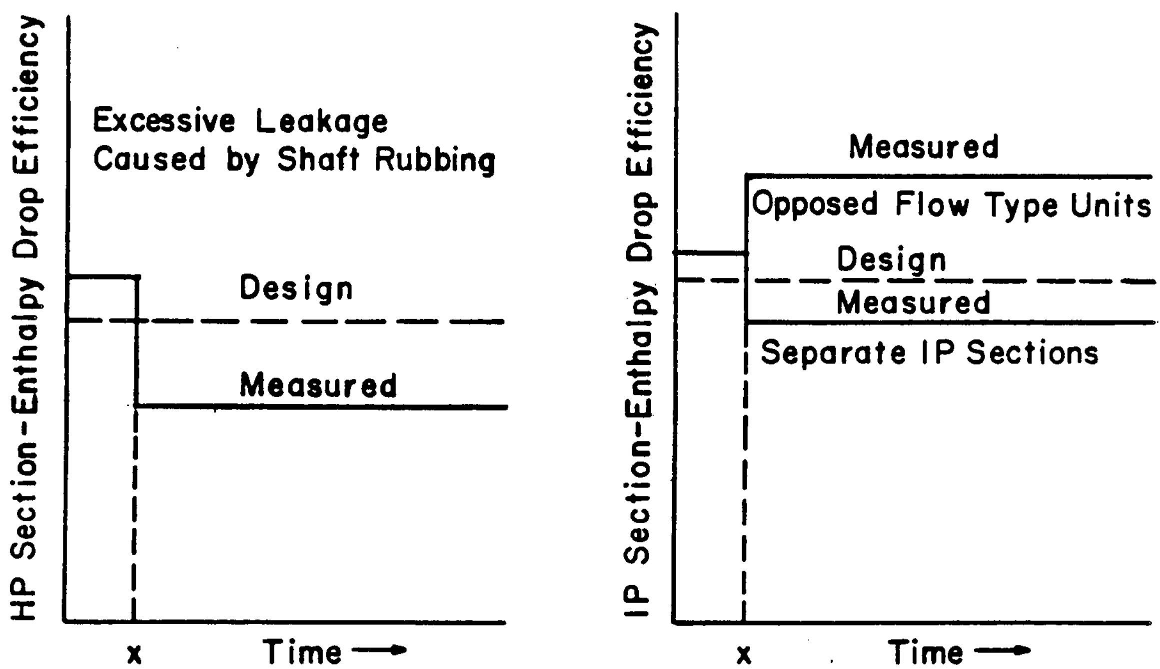

Figure 7-14

Supporting Information:

- o Was there a vibration problem at time X?
- o Was there a severe thermal problem at time X (cold start-up, water induction, load swing, temperature control problem, etc.)?
- o Has $\sqrt{P/V}$ or P at the first stage shell decreased for a given throttle flow?

Erosion - Solid Particle

This subject focuses on the damage done by oxide material which spalls off hot boiler and pipe surfaces, then, carried by steam, cuts and scratches away turbine material in high velocity zones. The immediate effects are to increase flow areas and decrease efficiency. This is in contrast to the closing effects caused by larger, more dense particles such as weld beads, as discussed later.

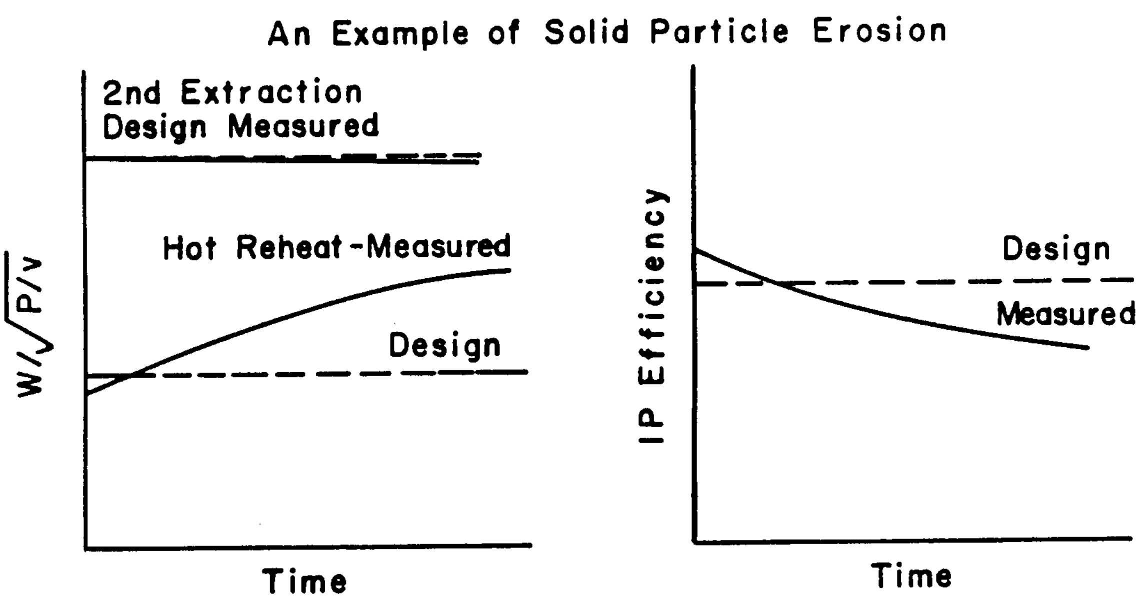

Figure 7-15

The above example indicates that it is generally the admission stage (i.e., the first high pressure or first reheat stage) where erosion effects are most severe. The damage declines at subsequent stages as the particles become finer while passing through the turbine and as they are drawn off at extraction points.

Sometimes the damaging capability of the fine oxide is revived at the crossover so that the first LP stage will be strongly attacked even though the last IP stage is unaffected.

Material differences between one stage and another (spillstrips, packings and nozzle materials, also physical size of nozzles), may also cause an unexpected local change in erosion damage. Softer materials or smaller profiles are more vulnerable to erosion damage.

Supporting information: a past history of erosion.

Note that both erosion and rubbing tend to increase flow areas. The key elements for differentiating which problem exists are (see Table 7-3):

- o Erosion occurs relatively gradually — rubbing is sudden.
- o Erosion can cause large effects in the IP section, P/V functions -- rubbing cannot.
- o Leakage can sometimes cause an apparent increase in IP efficiency (for units with HP and IB sections combined in one outer shell).

Internal Damage (not erosion type damage)

Internal turbine damage is commonly caused by:

- o Steam-carried hard particles from the boiler.
- o Turbine components which have broken or vibrated loose.
- o Foreign material left in the turbine or boiler.

In most cases the result will be reduced flow area and a loss of capacity.

An Example of Internal Damage

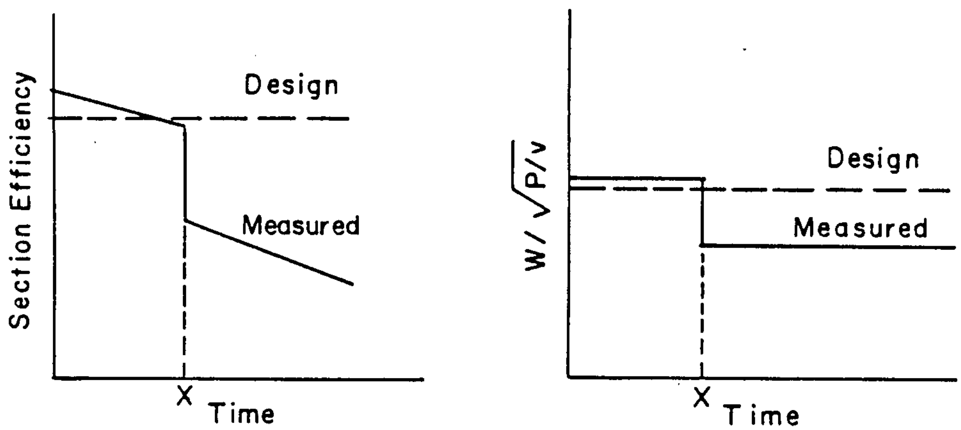

Figure 7-16

The hypothetical example in Figure 7-16 shows a probable damage situation at time X. Both efficiency and flow areas have been decreased by a sudden incident.

In addition to the pictured effects on efficiency and flow area, internal damage can be accompanied by vibration (temporary or permanent) and by a change in thrust. Not infrequently an incident will have secondary perturbations. The possibility of subsequent reliability loss is very difficult to appraise in the case of internal damage, but important to consider, especially where a vibration change indicates the rotating structure has changed.

Evidence of heavy erosion damage is occasionally a precurser to potential internal damage caused by tenon erosion and subsequent loss of bucket covers.

Deposits

Chemical deposits (oxides, sulfates, carbonates, silica, etc.) can heavily impact performance and capacity. In recent years, improved control of feedwater chemistry has generally decreased the magnitude of such losses. Still, losses of several percent for groups of stages is occasionally found. This is especially true where large amounts of reheat desuperheating flow are used, since the desuperheating water bypasses the boiler drum, carrying over more contaminants.

Extreme cases of copper oxide deposits in high pressure turbines have severely limited flow capacity and increased thrust to a condition of forcing a shutdown.

Deposits usually accumulate in the steam path in areas of pressure drop where steam conditions have been lowered to the point where the chemical saturation point is no longer exceeded. For impulse stages the deposition will be primarily in the stationary blades, on the suction surface, in the vicinity of the throat. In reaction stages, the deposit will accumulate in both rotating and stationary blades.

Figure 7-17 shows a hypothetical deposition case where both efficiency and flow areas have been reduced with time. Thrust may change, however vibration is usually not affected. Note that in double flow sections deposition may occur equally in both directions with no net change in thrust.

An Example of Steam Path Deposits

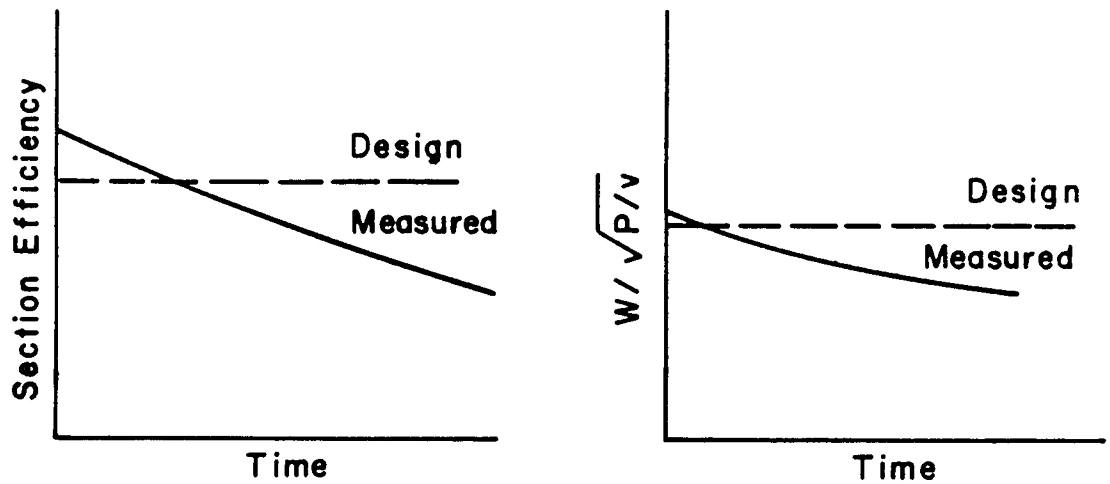

Figure 7-17

Some deposits are sensitive to temperature and may spall off during a shutdown or a large load swing. Some are also soluable in water and have been washed from the steam path by steam that includes water to either dissolve or break loose the chemical layer by impact. Water washing does not normally provide a lasting value and includes some risk.

Deposit presence can occasionally be confirmed by temporary improvements in efficiency and a rise in hotwell conductivity during transients or after a shutdown.

<u>Summary: Pre-outage Predictions for Morgantown 2</u>

At Morgantown 2, prior to the 1984 outage, historical test data were not available to permit rigorous application of the diagnostic chart techniques described above. This was not a severe handicap in this case since the presence of both erosion and seal wear could easily be determined from the available test data and previous test reports. Future operation will include the capability to plot, versus time, the critical functions of pressure, flow, $W/\sqrt{P/V}$, HP efficiency, IP efficiency and a variety of other turbine performance-related values.

The following predictions were made:

- Severe 1st stage erosion on the 1st valve arc nozzle (37% area increase)
- 2nd stage nozzle erosion up to 25% area increase
- Main steam piston ring leakage (estimated over 100,000 lb/hr)
- Excessive N2 leakage of 100,000 to 300,000 lb/hr
- Excessive clearance in the HP and IP packings and tip seals
- Severe erosion on the 1st reheat and 1st LP nozzles
- Moderate erosion in other IP nozzles
- Leakage flow (estimated) of 100,000 lb/hr from startup blowdown line to the condenser

Beyond the nine-step diagnosis program described in the preceding pages, two other calculating methods are occasionally useful. These are:

- Partial recovery of losses by following stages
- Determination of effects on heat rate due to changes in KW output.

Both methods are described in Appendix VI.

APPENDIX I

Steam Path Appraisal

With the turbine open for inspection, the high pressure and intermediate pressure sections were carefully examined for sources of loss.

Table 7-4 below summarizes the results of the inspection.

Table 7-4

HP Losses	Loss From New	Recoverable Losses
Tip Leakage	4335	4335
Packing Leakage	3860	3560
1st Stage Nozzle Erosion	3790	3790
Steam Path Roughness	1970	1670
N2 Packing Excess Leakage*	3060*	3060*
Total	17015	16415
Effect on Enthalpy Drop Effic.	8.0%**	7.7%**

*Does not effect enthalpy drop efficiency.
**If 117000 lb/hr piston ring leakage is included the known losses increase by 5500 kw and the efficiency loss by 2.6 points.

IP Losses		
Tip Leakage	1113	1113
Packing Leakage	600	500
8th Stage Nozzle Erosion	2300	2300
Steam Path Roughness	560	500
Total on IP Section	4573	4413
Effect on IP efficiency	3.0%	3.0%
LP Losses	875	875
Cycle Losses		
N2 Leakage Effect on RH Section	2800	2300
Total Unit	25258 (4.5%)	24058 (4.3%)
Effect on Heat Rate	332 (3.7%)	

Discussion of Results

- Excessive leakage due to seal rubs.

  Average packing clearance in the HP-IP sections were approximately 70 mils (tip seals were about 50% greater).

  Probable excess N2 leakage flow was calculated at 110,000 lb/hr. Added to the normal flow (52,000 lb/hr) this totals 162,000 lb/hr. This is high, but significantly less than the worst range estimated from the pre-outage test results. Excess 2nd stage packing leakage was 150,000 lb/hr.

- Leakage due to piston rings

  The number 1 and 2 inlet pipe piston rings were found stretched and eroded. The leakage area could not be accurately estimated, but the over 100,000 lb/hr assumed prior to the outage seemed reasonable.

- Solid particle erosion on nozzles

1st Stage	Approximate Increase
1st valve port	32%
2nd valve port	25%
3rd valve port	7%
4th valve port	0%
2nd Stage	+3%
8th stage (1st reheat)	+49%

- HP Section Efficiency

  When corrected for probable piston ring leakage losses, the test loss in this section appeared accurately explained.

- IP Section Efficiency

  Only 3% of the expected 4% IP loss was found by the appraisal. This has a good effect of reducing the amount of N2 leakage flow needed to explain the high apparent IP efficiency of the test.

- 1st State Shell Pressure

  The combination of excess N2 and 2nd stage diaphragm packing leakage would explain about 6% lowness of the 1st stage shell pressure. Estimated piston ring leakage and turbine bypass leakage would explain another 5%. Finally, the low throttle enthalpy (about 30 Btu) would explain 4.2% more, thus providing an apparently satisfactory understanding of the low 1st stage shell pressure.

Conclusions from Inspection

The pre-outage diagnosis and the inspection were in strong agreement. Only in the area of the N2 flow was there significant deviation from the predicted levels, and even it was within the range predicted.

APPENDIX II

Closing Steam Path Appraisal

Significant damage (seal rubbing, nozzle erosion, pockmarking, etc.) had been found when the unit was opened. It was deemed desirable to evaluate whether the extensive repair procedures had consistently achieved the quality assumed for the predicted recovery of los performance.

Results of Closing Appraisal

With the exception of items discussed below, the repairs had been skillfully completed:

- o Reconstruction of eroded and damaged nozzle profiles did not fully provide the intended quality.
  - some discharge edges were excessively thick
  - some profiles were suspected of significant deviation from design.
  - some surface finishes were poorer than desirable
- o Time did not permit improvement to the profile or edge thickness problems. The surface roughness condition was greatly improved by PEPCO blade specialists.

APPENDIX III

POST OUTAGE PERFORMANCE MEASUREMENTS AND ANALYSIS

These results represent a basis for determining the success of the effort to establish a high level of performance for the turbine cycle. The same procedure used at the pre-outage analysis was used. Note that these data will also be used for benchmark conditions from which deviations with time will be recognized and followed.

Step 1: Flow Capacity

VWO Design Flow	=	3,964,071
With normal 2.5% excess	=	4,063,173
Test VWO flow (corrected)	=	4,242,966

This is 4.42% high

Questions:

- o Has nozzle erosion already occurred?
- o Is first stage shell pressure low?
- o Is there still some flow bypass?

Step 2: Throttle $W/\sqrt{P/V}$ Across the Load Range

Test data were not available at the first valve point (#2 valve crack point) and heat balance information near the #3 valve crack point was not believed correct for the valve point condition. Figure 7-18 shows the test data and design information. Only the VWO point is believed fully trustworthy.

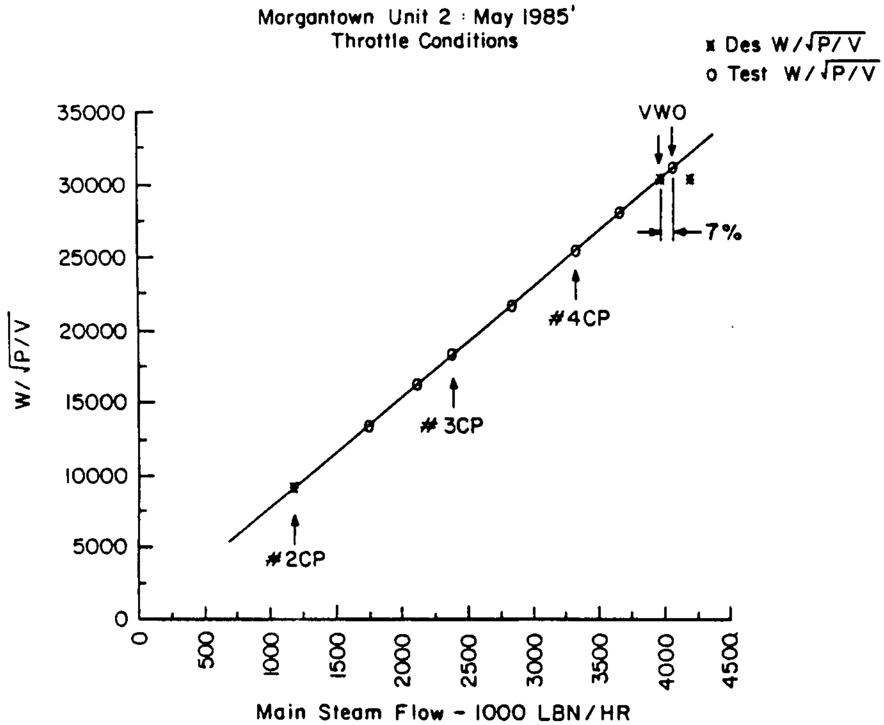

Figure 7-18

Using the recommended relationship at VWO:

$$\frac{A_{test}}{A_{design}} = \frac{W_{test} \times N_{des}}{W_{des} \times N_{test}} = \frac{0.9}{0.9} \times \frac{4242966}{4063173} = 1.044$$

The 4.4% extra area, if confined to the 1st arc would suggest it has increased by about 18%. It is surprising that signs of heavy erosion appeared so quickly.

Step 3: VWO Flow/1st Stage Pressure Analysis

Figure 7-19 shows the pressure/flow results.

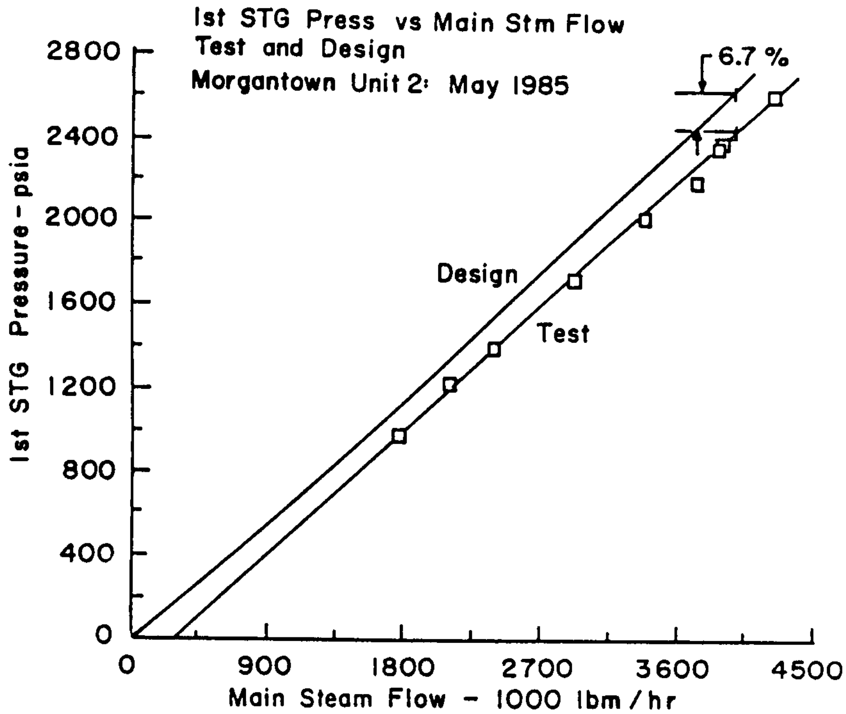

Figure 7-19

Significant improvement has occurred. Pre-inspection tests showed this pressure to be low by 17%; now it is low by 6.7%. But even 6.7% lowness is a source of concern. Several questions need consideration:

- o Can there still be a bypass or a piston ring leakage [seems unlikely]?
- o Can there be a flow measurement error [new calibrated flow nozzle in place -- very unlikely].
- o Can there be excess N2 or 2nd stage diaphragm packing leakage? Extra clearance was provided in these seals -- probable effect would be to drop the pressure about 2.2%.
- o Can there be a pressure measurement error? Note that the test line does not extrapolate to zero pressure at zero flow.

- o Can there be a fixed flow leakage around the flow nozzle (see the later discussion concerning the outboard seal on the BFP).

Step 4: $W/\sqrt{P/V}$ Analyses at HRH, Crossover and Extraction Points

Location	$W/\sqrt{P/V}$ Design $x10^{-3}$	$W/\sqrt{P/V}$ Test $x10^{-3}$	(Test-Des) x100 / Design
HRH Bowl	166	177	+6.6%
#2 Extraction	323	331	+2.7%
Crossover	475	475	0%
#4 Extraction	765	809	+5.8%
#5 Extraction	2280	2358	+3.4%

The consistent pattern of highness in the test results is a hint that the calculated flow through these stages is slightly exaggerated. Perhaps the spray flow is exaggerated.

Step 5: Compare Test and Design HP Section Efficiency Across the Load Range

Figure 7-20, below, shows test and heat balance values for the HP section.

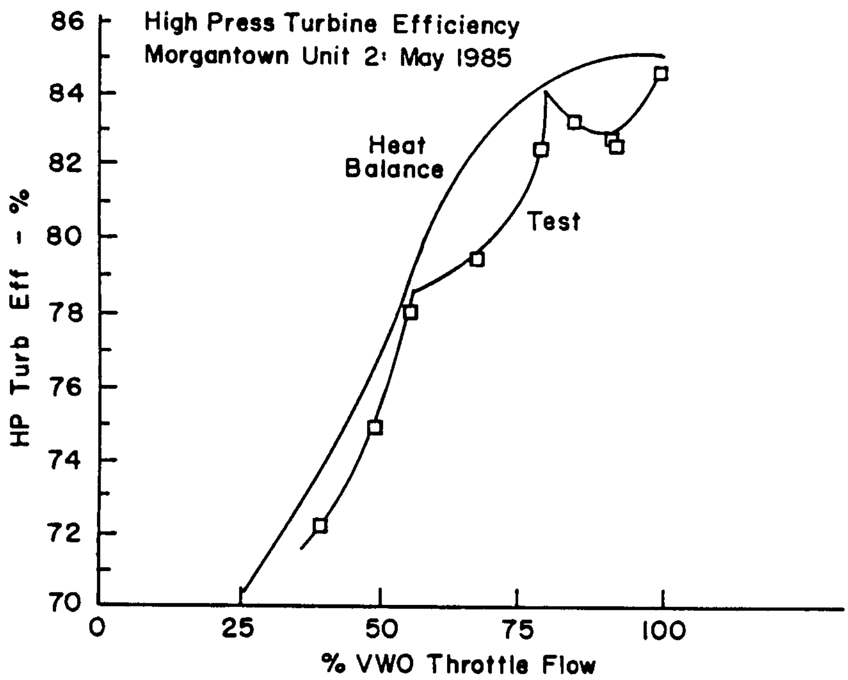

Figure 7-20

The VWO point is about 0.5% lower than design (and about 3% below the best expected, considering about 2.5% manufacturers margin).

The dashed line between test points are a rough interpretation of the valve loops. If properly drawn, it suggests that the deviation from design is fairly constant for #2, #3 and #4 valve points.

It had been hoped that the HP section would return to service with better than heat balance performance -- perhaps 86.5% efficiency. Based on previous sections, it appears that some erosion, some rubbing (low 1st stage shell pressure) and some incomplete nozzle repairs -- thick edges and marginal profiles -- have prevented a full recovery.

Still, this level of performance is a strong improvement over past levels and one which all parties are pleased.

Step 6: Compare Test and Design IP Section Efficiency Across The Load Range

Figure 7-21 shows the corrected test results for the IP section.

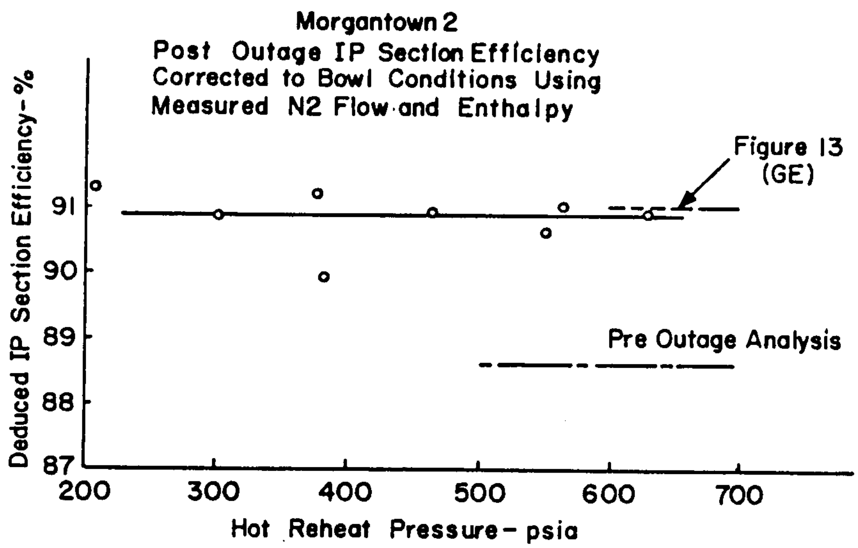

Figure 7-21

This appears to be a highly satisfactory result, both for level and magnitude of improvement. It is noted that these results are based on measured N2 flow and enthalpy using the N2 blowdown valve bypass. The level and flatness of the resulting efficiency indicate the system is providing accurate measurements.

Step 7: VWO Kilowatt Check - Post-Outage Test

Test Kw	-	628030
Test Flow	-	4224211
HB Kw	-	597556
HB Flow	-	3964071

Kw Corrections to HB Conditions:

HP Efficiency low	-815
HP flow high	+5895
IP flow high	+41003
Mixed IP bow enthalpy is high	+2476
Back Pressure is high	-2378
Exhaust Loss is high	-3400
RHTR ΔP is low	+4780
Air preheater flow is low	+5647
N2 flow is high	-1344
Total corrections	+51864

Expected output (HB + Correction)	649420
Actual Test Output	628030
Unexplained Difference	21390 (3.3%)

The above Kw check is not good. It suggests that 3.3% greater Kw are expected than measured. This is far too much than can realistically be assigned to poorness in the LP section. Based on previous inspections, 1% is the most overall loss that could be blamed on that section.

Questions to be considered:

- o Is the flow measurement exaggerated?
- o Is there an isolation problem?
- o Is the electrical output measurement in error?
- o Is the RH spray flow measurement exaggerated?
- o Does the BFP Tb require excessive flow not properly measured?

In regard to these questions, test nozzle flow measurement is known to be accurate. One possible source of an isolation problem is the unmeasured BFP seal leakage from the outboard seal. This could cause the calculated feedwater to be in error by the amount of seal leakage from the pump. The electrical output measurement is believed to be the best available and highly accurate. The BFP Tb flow is believed to be at most, a small source of error. The reheat spray flow is suspect and will be investigated.

Note the predominance of indications suggesting the turbine stage flows are exaggerated:

VWO flow capacity

1st stage shell pressure

$W/\sqrt{P/V}$ at the HRH, #2 Extraction, #4 Extraction and #5 Extraction

Kw check

Step 8: Other Considerations

The above analyses indicate a generally satisfactory result from the tests, analyses, inspection repair and retest. Only two areas of real concern are apparent:

- Is erosion rapidly enlarging the 1st stage and 1st RH stage nozzles? Should alternative methods of operation be considered?
- Is there an isolation or measurement problem that can cause the perceived turbine flow to be exaggerated? A flow measurement for the BFP outboard seal is planned; the large reheat spray flow measurement will be checked.