Practical Application of Methods to Determine Leakage/Between HP and IP Turbines

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ABSTRACT

Accurate calculation of turbine efficiencies is the prime motivation behind performance testing. However, one area which has a substantial impact on calculated IP turbine efficiency is often neglected. This is the internal leakage from HP to IP on opposed flow turbines, commonly referred to as "N2 Packing Leakage". Two methods to quantify N2 leakage were presented by Booth and Kautzman in the paper "Estimating the Leakage from HP to IP Turbine Sections." Our paper presents the practical application of the Booth and Kautzman methods in addition to an analysis of the effect that N2 leakage has on calculated IP and LP turbine efficiencies, generation and heat rate. The results of N2 leakage tests performed on ten pulverized coal fired units ranging in size from 90 to 450 megawatts help to illustrate our testing and analysis procedures. This is an expanded version of "Analysis of Leakage Between HP and IP Turbines Using PEPSE®", presented at the 1990 Users Group Meeting.

N2 PACKING LEAKAGE

Leakage from the high pressure to the intermediate pressure section of an opposed flow turbine through the number 2 packing (N2 packing) is not routinely measured during turbine performance tests. In test data analysis, a design packing flow coefficient is often used in conjunction with the simplified Martin's formula (Reference 2), to predict N2 leakage as a function of measured first stage pressure and specific volume. If the N2 packing has sustained no wear or damage, and is performing according to design specifications, the design flow coefficient will provide a good estimate of N2 leakage. In most cases, however, the packing will begin to degrade immediately after start up and actual N2 flow can substantially exceed design. This increased leakage flow has significant effects on unit performance and operating costs.

EFFECTS OF N2 LEAKAGE

To understand the effects of N2 leakage, it is first necessary to have an understanding of the N2 flow path in relation to the turbine main steam flow. Figure 1 is a schematic of the major flows through the HP and IP turbines, including the N2 leakage flow path. On an opposed flow turbine, the HP and IP sections share a common shaft, with main steam entering the HP and hot reheat steam entering the IP at mid-span. After main steam expands through the first stage, it is separated from the hot reheat steam by the labyrinth type N2 packing. Because of the large pressure differential (approximately 1400 psi on a 2400 psi machine), there is a natural flow from higher pressure to lower pressure.

N2 leakage affects test results in many areas of the turbine cycle. The most significant impacts are on calculated values of IP efficiency, LP efficiency and gross turbine cycle heat rate. An unaccounted increase in N2 leakage will result in an increase in calculated heat rate, an increase in calculated IP turbine efficiency, and a decrease in calculated LP turbine efficiency.

First, increased N2 leakage decreases the mass flow through the HP turbine downstream of the first stage. This causes a

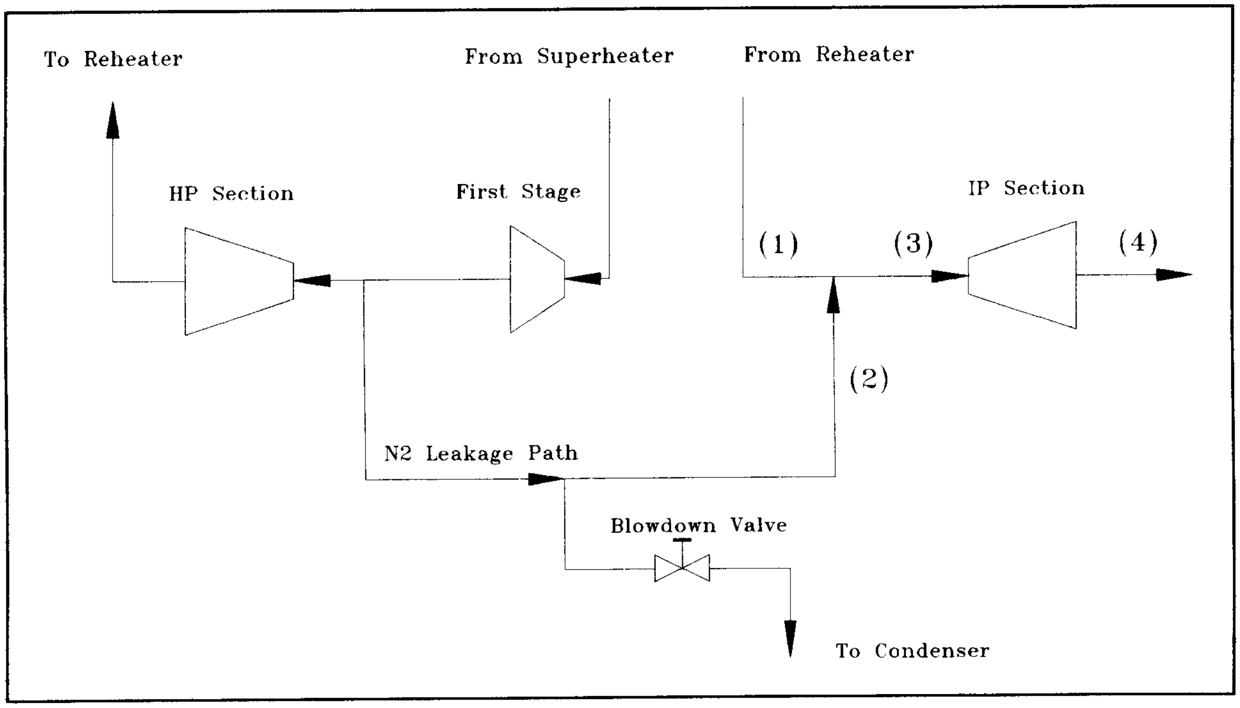

Figure 1 -- Leakage Flow Path

decrease in wheel power produced by the HP section. To calculate LP turbine efficiency in a unit with a two phase endpoint, HP and IP wheel power are subtracted from the measured generation, to yield required LP wheel power. If an increase in N2 leakage is not accounted for, a higher than actual HP wheel power would be calculated. Therefore, a lower LP turbine wheel power would be required to produce the measured generation, which in turn would cause a lower than actual calculated LP turbine efficiency.

Next, increased N2 leakage has a dramatic effect on calculated IP turbine efficiency. Hot reheat pressure and temperature are measured at location (1) in Figure 1, allowing the calculation of the enthalpy at (1). The enthalpy at (2) is the same as first stage enthalpy, however, the enthalpy at (3) can only be calculated if the flows at (1) and (2) are known. Location (4) is the crossover between the IP and LP turbine sections and this enthalpy is also calculated from direct pressure and temperature measurements. Typically, enthalpy drop tests will ignore the cooling effect of N2 leakage, calculating IP efficiency as: $(h_1-h_4)/(h_1-h_{4s})$, referring to the locations in Figure 1. Calculated this way, as the N2 flow increases, the cooling effect results in a lower crossover enthalpy and an increase in calculated IP turbine efficiency.

An accurate calculation of IP turbine efficiency depends on knowledge of the enthalpy at locations (3) and (4), therefore the flows at (1) and (2) must be obtained. Since these flows are interdependent, if either flow can be obtained, the second flow is easily calculated. The following example illustrates the importance of N2 leakage in the IP efficiency calculation. These calculations use valves wide open design pressures and temperatures to calculate IP turbine efficiency. The first calculation assumes a 25000 lbm/hr N2 leakage flow, while the second uses 50000 lbm/hr.

Define IP Turbine Efficiency:

$$\text{Efficiency} = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

Where: h_3 = enthalpy at location (3)
h_4 = enthalpy at location (4)
h_{4s} = isentropic enthalpy at location (4)

With 25000 lbm/hr Leakage:

T_1 = 1000°F
P_1 = 515.6 psia
h_1 = 1519.9 Btu/lbm
m_1 = 1687608 lbm/hr

h_2 = 1438.2 Btu/lbm
m_2 = 25000 lbm/hr

$m_3 = m_1 + m_2$

T_4 = 650.8°F
P_4 = 123.7 psia
h_4 = 1353.5 Btu/lbm
$m_4 = m_3$
h_{4s} = 1328.8 Btu/lbm

Calculate h_3: $h_3 = (m_1 h_1 + m_2 h_2)/(m_1 + m_2) = 1518.7$

$$\text{Efficiency} = \frac{1518.7 - 1353.5}{1518.7 - 1328.8} = 0.8699$$

With 50000 lbm/hr Leakage:

$T_1 = 1000°F$
$P_1 = 515.6$ psia
$h_1 = 1519.9$ Btu/lbm
$m_1 = 1662654$ lbm/hr

$h_2 = 1438.2$ Btu/lbm
$m_2 = 50000$ lbm/hr

$m_3 = m_1 + m_2$

$T_4 = 650.8°F$
$P_4 = 123.7$ psia
$h_4 = 1353.5$ Btu/lbm
$m_4 = m_3$
$h_{4s} = 1325.7$ Btu/lbm

Calculate h_3: $h_3 = (m_1h_1 + m_2h_2)/(m_1 + m_2) = 1517.5$

$$\text{Efficiency} = \frac{1517.5 - 1353.5}{1517.5 - 1325.7} = 0.8551$$

The resulting change in calculated IP turbine efficiency of 1.5 percent illustrates that unaccounted increases in N2 leakage could mask significant degradation in the IP turbine or provide a false indication of increasing IP turbine efficiency.

The masking effect on IP turbine efficiency also has an effect on calculated LP turbine efficiency. Because the IP inlet enthalpy is incorrect, the calculated IP wheel power is also incorrect, yielding a similar effect on LP efficiency as the flow error in the HP turbine section. The apparent increase in IP wheel power decreases the required output of the LP section and, therefore, reduces the calculated LP turbine efficiency.

The following calculations illustrate the effect of N2 leakage on the calculation of gross turbine cycle heat rate. The first calculation is the correct calculation of the gross turbine cycle heat rate of a 448 Mw unit. These are the design conditions, including the N2 leakage. The second calculation assumes that N2 leakage is at design when it was actually two times design. Note that all of the parameters are the same as in the first equation, except that generation is down by about 1.5 Mw. The calculated gross turbine cycle heat rate is approximately 27 Btu/KwHr higher than in the design case. The third calculation results in the actual turbine cycle heat rate and properly accounts for the increased N2 leakage. This shows that the gross turbine cycle heat rate actually increased 13 Btu/KwHr from design by doubling design N2 leakage, and not properly accounting for the increased leakage yields an unaccountable loss of 14 Btu/KwHr. Note that the only difference between the second and third heat rate calculation is the change in reheat flow. From these calculations it is evident that quantification of N2 flow is critical for accurate plant performance indication.

Design Turbine Cycle Heat Rate:

$$GTCHR = 7926 = \frac{3,034,463(1456.3-471.2)+2,717,141(1518.8-1311.1)}{448435}$$

Turbine Cycle Heat Rate without N2 leakage accounted for:

$$GTCHR = 7953 = \frac{3,034,463(1456.3-471.2)+2,717,141(1518.8-1311.1)}{446788}$$

Turbine Cycle Heat Rate without N2 leakage accounted for:

$$GTCHR = 7939 = \frac{3,034,463(1456.3-471.2)+2,682,351(1518.8-1311.1)}{446788}$$

METHODS TO QUANTIFY N2 LEAKAGE

Two methods, blowdown and temperature variation, are currently accepted for evaluation of N2 packing leakage using existing plant hardware. Each method has unique advantages and disadvantages, but both require the disruption of normal plant operation.

Blowdown Method

The blowdown method utilizes the emergency blowdown valve to divert the N2 leakage from the IP turbine inlet to the condenser. The emergency blowdown valve is a safety mechanism designed to prevent turbine overspeed by removing HP turbine leakage steam and passing it directly to the condenser, bypassing the entire IP and LP turbine sections. According to Booth (Reference 6), the flow passing capability of the blowdown valve should be more than sufficient to remove all of the N2 leakage from the turbine and route it to the condenser. This would make the energy at location

(1) in Figure 1 equal to the energy at location (3). By eliminating all of the flow at location (2), the actual IP turbine efficiency can be calculated directly. The IP efficiency is then used with data collected when the blowdown valve is closed to calculate the N2 leakage flow.

Blowdown Test Data and Analysis

A minimal data set is required for the blowdown test. Required data are hot reheat temperature and pressure, crossover temperature and pressure, and data to calculate first stage enthalpy. Two 30 minute test runs at a single valve position, the first with the blowdown valve closed, the second with the blowdown valve open, are required. ASME PTC-6 stability requirements must be met for accurate results.

The first step in data analysis is the calculation of IP turbine efficiency using the blowdown valve open data. Calculate the IP turbine enthalpy drop efficiency using the ASME PTC-6 method. If all N2 leakage is diverted through the blowdown valve, this will yield an actual IP turbine efficiency for the current valve position. Next, the data set with the blowdown valve closed is used. Using the IP efficiency from the blowdown open case, the blowdown closed inlet enthalpy can be calculated. The mass fractions of N2 leakage and reheat flow can then be calculated iteratively.

Results of Blowdown Tests

We have run blowdown tests on three different units, with very consistent results. Unfortunantly, the blowdown results did not replicate the results of the temperature variation tests. In fact, the calculated leakages from the blowdown tests do not vary far from design, as can be seen from Table 1. It should be noted that the blowdown tests on Unit 1 and Unit 2, which are sister units, calculate very similar leakage flows. Further analysis indicates that the blowdown valves were flow limiting, allowing only slightly more than design N2 leakage to pass through the blowdown valve. Calculations of the maximum flow passing capability of the Unit 3 blowdown valve resulted in a maximum flow of approximately 2.4 percent of first stage flow. The consistency between the maximum flow calculation and the blowdown test result confirmed that the blowdown valve was flow limiting. Therefore, the results of blowdown test run on this unit was invalid.

Table 1 -- Blowdown Test Results

UNIT	DESIGN LEAKAGE	BLOWDOWN RESULTS
Unit 1 - 165 Mw	2.2 %	1.3 %
Unit 2 - 165 Mw	2.2 %	1.2 %
Unit 3 - 285 Mw	1.9 %	2.4 %

If successful, the blowdown test yields the true IP turbine efficiency. N2 leakage can be calculated directly from test data, but is dependent on first stage enthalpy. If the blowdown system is capable of passing the entire N2 flow, direct calculation of IP efficiency is an advantage. Before attempting to run a blowdown test, the flow passing capability of the blowdown valve should be calculated to see if it is sufficient to pass the expected N2 leakage.

Temperature Variation

The temperature variation method uses the difference between the enthalpy at the first stage of the HP turbine and the enthalpy at the IP turbine, upstream of the intercept valve, to estimate N2 leakage. Because of the lower enthalpy of the N2 leakage from first stage, there is a cooling effect on the steam at the IP turbine inlet, which carries on down stream to crossover. This effect is maximized by decreasing throttle temperature, and minimized by decreasing hot reheat temperature. Mathematically, this method assumes that IP turbine efficiency remains constant for a fixed valve position. The two data sets, collected under different boundary conditions, result in two independent equations and two unknowns, IP efficiency and N2 leakage. Two equations and two unknowns yield a unique solution for IP efficiency and N2 leakage. The problem can also be solved graphically, as discussed in the next section. The temperature variation method accounts for N2 flow and any other leakages between the HP and IP turbines, such as snout ring leakage and leakage through the turbine case. These latter leakages are rare, however they should be kept in mind.

Temperature Variation Test Data and Analysis

Data required for the temperature variation test are hot reheat temperature and pressure, crossover temperature and pressure, and data to calculate first stage enthalpy. As a minimum, two test runs of one hour each are required. ASME PTC 6 stability requirements should be met or exceeded. To run a temperature variation test, the hot reheat and superheat temperatures are set to a temperature differential of approximately 50°F (i.e. hot reheat = 1000°F, superheat = 950°F). Test data is collected and IP efficiency is calculated using assumed values of N2 leakage of 0 percent and 10 percent of first stage flow. The assumed leakages of 0 and 10 percent are arbitrary and were chosen because they will usually bound the actual leakage value. The results of these calculations are plotted as an IP efficiency vs. leakage flow, yielding a straight line (Figure 2, Depressed SH Temp). Next, the unit is set up with a 50°F temperature differential in the opposite direction (i.e hot reheat 950°F and superheat = 1000°F). Another set of test data is collected and IP efficiencies are calculated, again using the same assumed range of N2 leakages. The results are plotted on the same graph (Figure 2, Depressed RH Temp). The intersection of these lines indicates the true HP to IP leakage and the true IP turbine efficiency. In many cases it may not be possible to achieve a 50°F variation in both directions. For best results, the maximum temperature differential within the turbine manufacturers warranty constraints should be obtained.

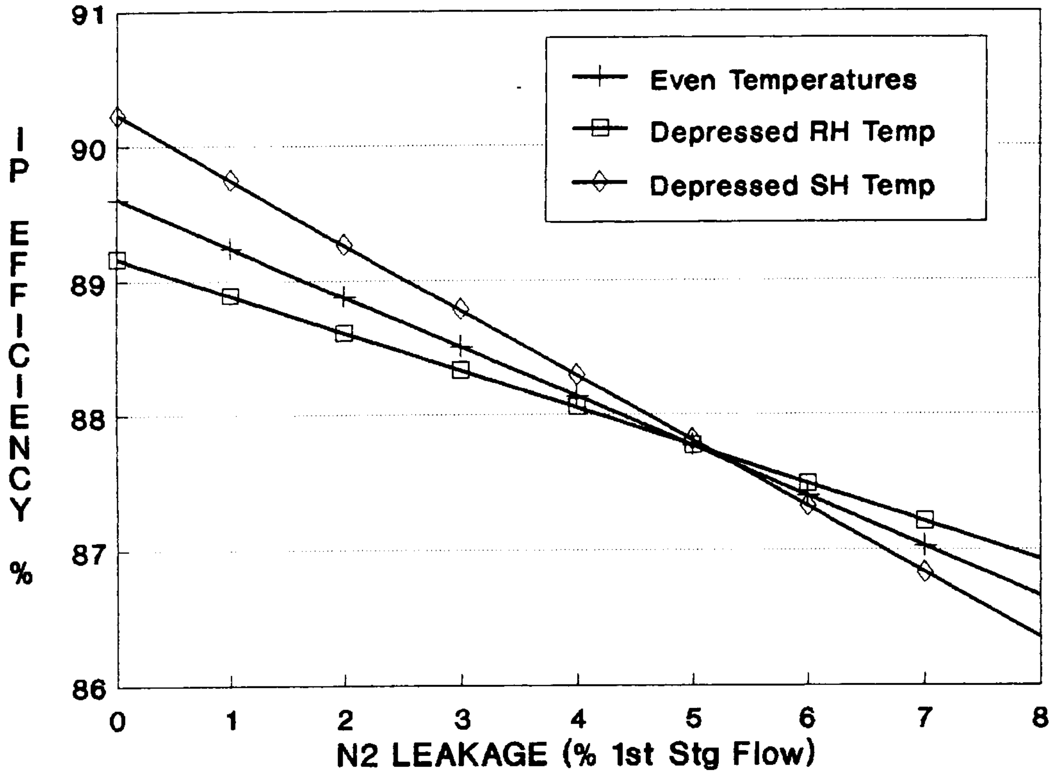

Figure 2 -- N2 Leakage vs. IP Efficiency

Results of Temperature Variation Tests

We have successfully run temperature variation tests on ten units since beginning to analyze N2 leakage. Our studies indicate the results of the tests are sensitive to several items. Initially, there were concerns about sensitivity of the results to superheat sprays, throttle valve position, make up flow, continuous blowdown, unit stability, and first stage enthalpy. Studies based on PEPSE® test data models indicate that unit set up has no significant effects, provided that a consistent set up is used. Figure 3 illustrates the negligible effect of accounting for superheat sprays, make up flow, and continuous blowdown. Throttle valve position and pressure drop across the throttle valve are not significant in the N2 leakage calculation, provided valve position is constant over a single series of temperature variations. The two most important parameters in consistently determining N2 leakage and correct IP turbine efficiency are unit stability and first stage enthalpy.

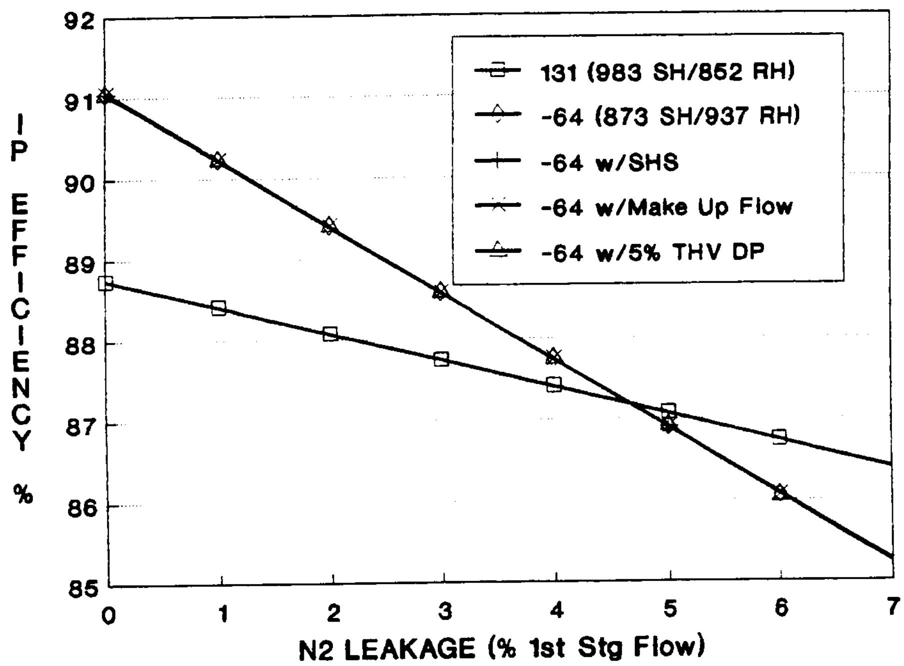

Figure 3 -- Unit Set Up Sensitivity - 150 Mw

Unit Stability

Figure 4 is a stability graph of throttle temperature, throttle pressure and hot reheat temperature vs. time for a depressed reheat temperature run on Unit 3. The test was run at 220 Mw. Throttle temperature was stable for the entire 60 minutes, but throttle pressure and hot reheat temperature did not stabilize until between 20 and 30 minutes. To analyze the effects of unit stability, we separated this one hour data set into two 30 minute data sets. In Figure 5, Test 1 corresponds to the first 30 minutes and Test 2 represents the second 30 minutes. Test 5 is a stable, depressed throttle temperature test at the same control valve position. In this case, the difference between stable and unstable test results is about 0.5 percent in efficiency and 0.5 percent in leakage. As a general rule, ASME PTC 6 stability requirements should be met, and exceeded, if possible.

When it is not possible to achieve a 50°F temperature variation in each direction, unit stability becomes even more critical. The reduced temperature differences result in lines of a more similar slope, and therefore, any instability which shifts either line will result in greater errors in indicated N2 leakage.

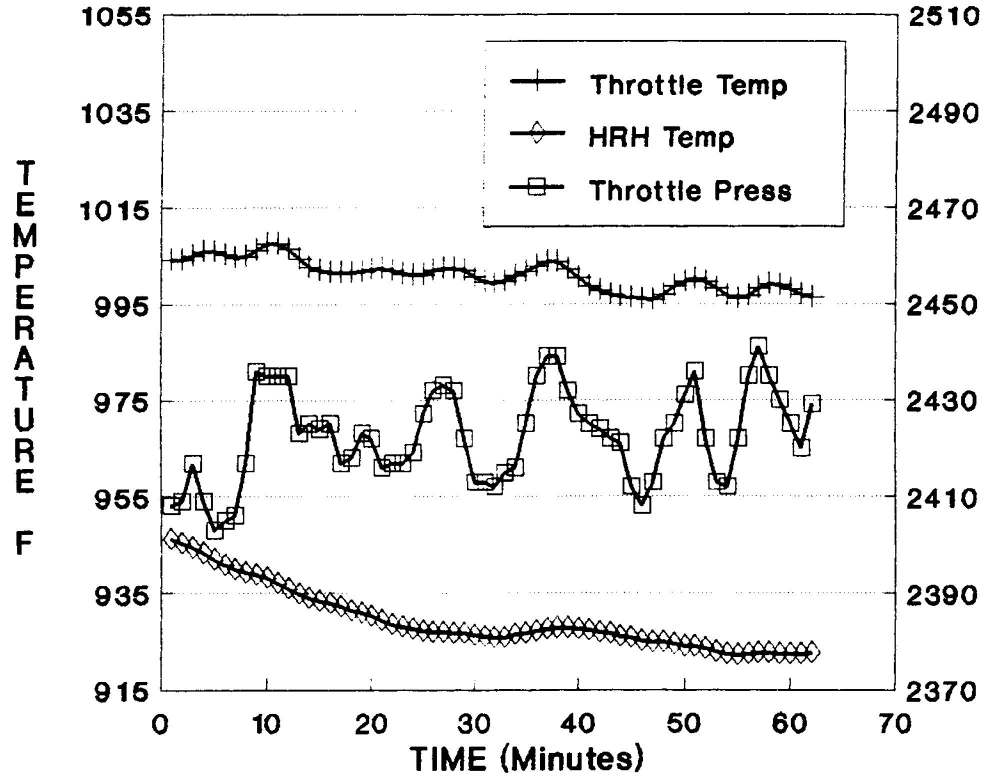

Figure 4 -- Test Stability - 220 Mw

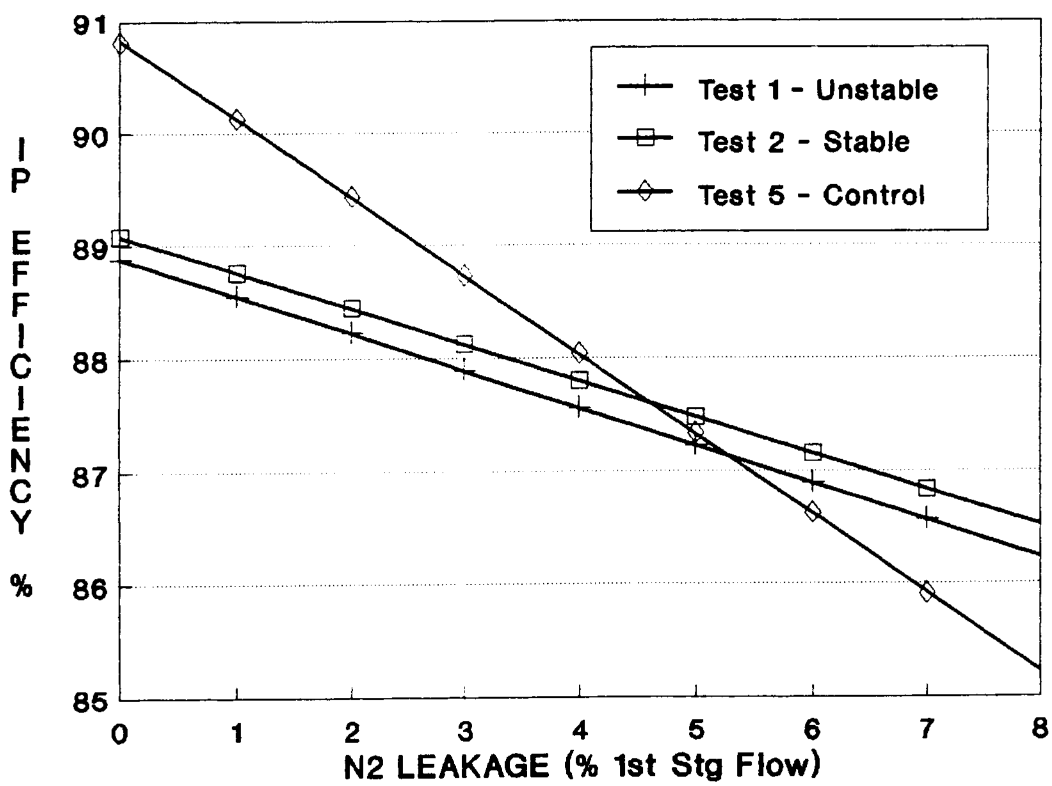

Figure 5 -- Test Stability Sensitivity - 220 Mw

First Stage Enthalpy

Determining first stage enthalpy can be a difficult proposition because most turbines have no provisions for measuring first stage temperature. Using only the measured first stage pressure, a number of methods are available to estimate first stage enthalpy. Design first stage efficiency and measured pressure will yield one enthalpy, while assuming a straight expansion line from the governing stage inlet to high pressure turbine exhaust (cold reheat) and the measured pressure will yield a slightly different enthalpy. Figure 6 depicts the difference between results using the straight expansion line and design first stage efficiency. Note that although the IP efficiency shifts about 0.25 percent between the two different methods, the associated leakage is virtually identical. Further analysis indicates that N2 leakage calculated by the temperature variation method is not very sensitive to the method used to determine first stage enthalpy. As long as a consistent method for determining first stage enthalpy is employed, the IP efficiency degradation can be trended.

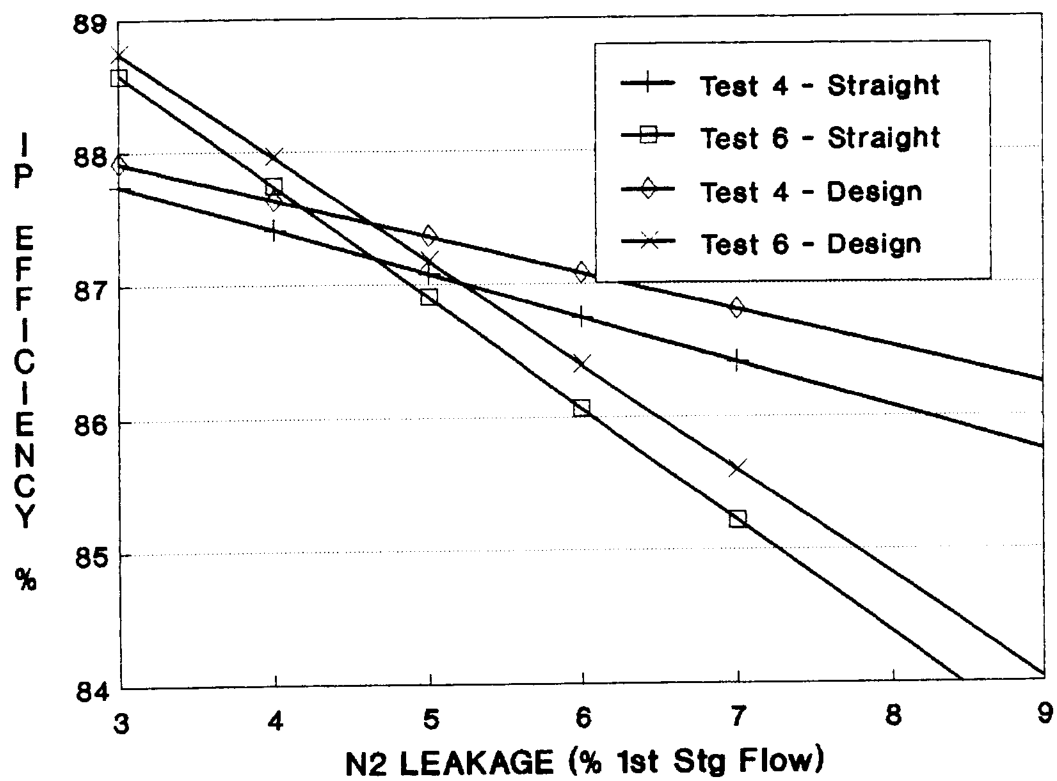

Figure 6 -- 1st Stage Enthalpy Sensitivity

Test Results

Figures 7, 8 and 9 present the results of a series of N2 leakage tests on a 285 Mw GE unit. Three sets of temperature variations were run on three different days, with very consistent results. All three loads indicate slightly less than five percent N2 leakage. The results of temperature variation tests from ten different units are presented in Table 2. Note that the leakages range from two times design to almost five times design, and the heat rate and generation effects are very significant.

Unit 8 is a 447 Mw GE unit which was due for a major turbine outage in Spring 1991. Three sets of tests run on Unit 8 over a nine month period very repeatably indicated N2 leakage of 3.0 percent of reheat flow. Results of the steam path audit performed during the outage matched the results of the N2 leakage analysis, indicating a flow area of approximately two times design. This is further verification of the temperature variation method.

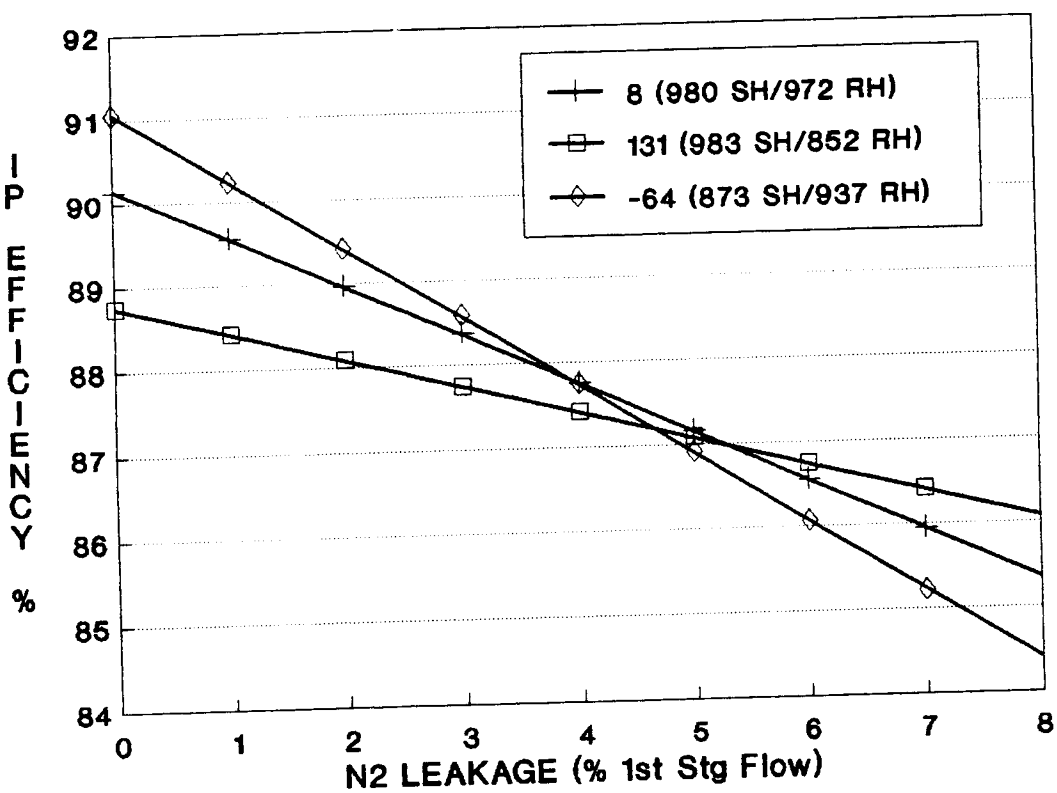

Figure 7 -- Unit 4 N2 Leakage - 150 Mw

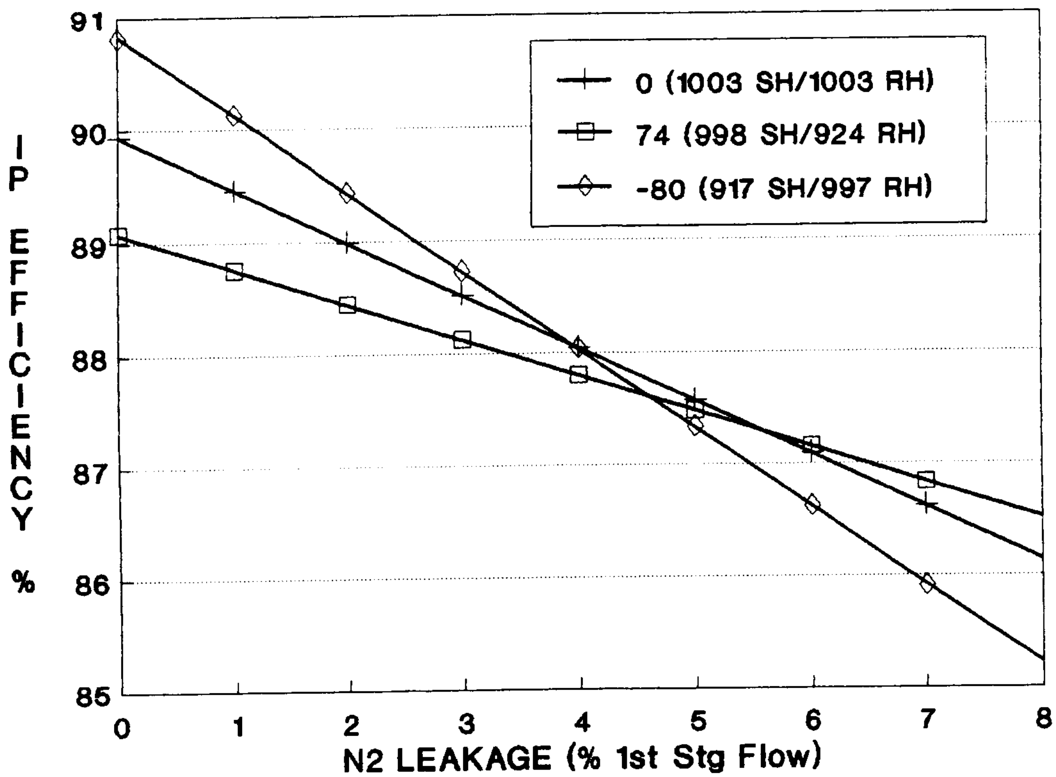

Figure 8 -- Unit 4 N2 Leakage - 220 Mw

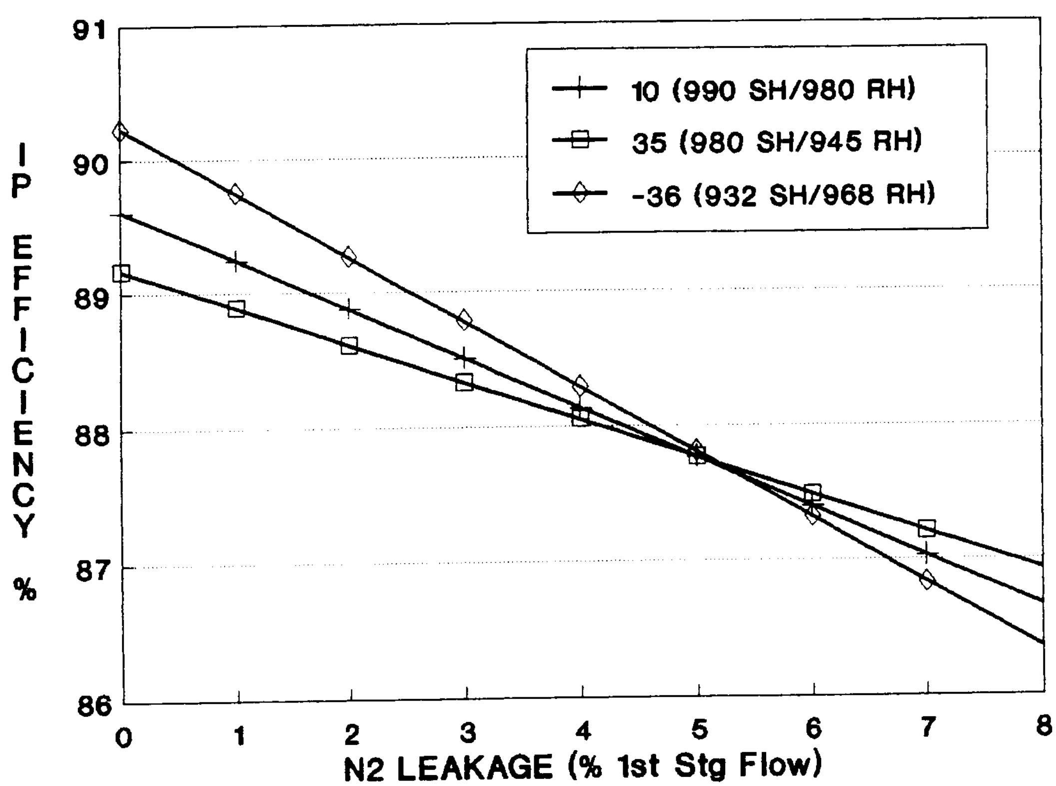

Figure 9 - Unit 4 N2 Leakage - 285 Mw

Table 2 -- Temperature Variation Results

Unit	Description	Design Leakage (%)	Tested Leakage (%)	IP Effic W/Design Leakage (%)	IP Effic W/Tested Leakage (%)	Heat Rate Cost Leakage (Btu/KwHr)	Generation Cost Leakage (Mw)
1	165 Mw GE	2.2	10.0	86.6	84.4	*	*
2	165 Mw GE	2.2	8.0	87.7	86.3	*	*
3	285 Mw GE	1.9	8.0	90.3	88.3	53	4.9
4	285 Mw GE	1.9	5.0	89.0	87.8	26	2.5
5	285 Mw GE	1.9	4.0	*	*	16	1.6
6	90 Mw GE	2.4	13.0	*	*	*	*
7	90 Mw GE	2.4	13.0	*	*	*	*
8	447 Mw GE	1.4	3.0	88.6	87.9	15	1.5
9	447 Mw GE	1.4	4.5	87.1	85.5	38	4.5
10	427 Mw GE	1.4	5.0	87.4	85.6	26	2.8

* Not Available

Economic Analysis of N2 Leakage

Unit 9, a 447 Mw GE unit was tested in November 1990, then retested in March 1991. At each test, N2 leakage was approximately 4.5 percent of reheat flow, which is three times the design N2 leakage flow of 1.35 percent. Heat balance calculations yielded an increase in turbine cycle heat rate of 38 Btu/KwHr, and a decrease in maximum generation of 4.5 Mw. Considering only the increase in fuel, the annual increase in operating cost is $142,000. This cost is based on $1.06/million Btu, 89 percent boiler efficiency, and 80 percent capacity factor. The payback period on replacing the N2 seals on this unit is less than one year.

Correcting Excessive N2 Leakage

Excessive N2 leakage can only be corrected during a major turbine outage. The seals can be replaced by standard packings or by the newer variable clearance packings. The practice in recent years has been to increase clearance on the standard packings to prevent rubs during startup and unit trips. This typically increases design leakage by a factor of two, causing an increase in heat rate and a decrease in generation. The variable clearance packings allow excess clearance during startup, then close down to design clearance at load. Theoretically, this will provide design leakage once the packings close. Tests of both types of packings immediately after a major outage provided some interesting results.

Unit 5, a 285 Mw GE unit, was tested extensively after replacing the N2 packings with the standard GE design. The clearances were also set to original GE specifications. Twenty-eight test runs of 30 minute duration were run over a period of three weeks. Prior to and during the test period, the unit either tripped or was shut down several times. It is suspected that the packings were rubbed during these shut downs and associated restarts, however, the test results indicated that the N2 leakage was between 2.5 and 3 percent of first stage steam flow at the end of the test period. Given a design N2 leakage of approximately 1.9 percent of first stage flow, the new standard GE design N2 packings resulted in a very near design steam leakage. If the tests had been run prior to any unit trips or turbine shutdowns, the leakage probably would have been even lower.

Unit 4, a sister unit to Unit 5, was tested both before and after a major turbine outage. As with Unit 5, the packings were replaced with standard GE packings set at design clearance. The pre-outage N2 leakage was 8 percent of reheat flow. Post-outage testing began immediately after start up, and early indications were that the N2 leakage was very close to design. However, as testing progressed, numerous unit trips and start ups occurred. Each time the N2 leakage test was run the leakage had increased. At the end of the test period, leakage was approximately 6 percent of reheat flow. This degradation occurred within approximately two months after start up.

Unit 8, a 447 Mw GE unit, was tested both before and after a major turbine outage. Based on the pre-outage leakage of 3.0 percent of reheat flow, an economic analysis justified replacing the standard N2 packings with variable clearance packings. If the design N2 leakage of 1.4 percent of reheat flow could be achieved, the payback based on fuel savings would be two years. The unit was tested three weeks after completion of the outage, and N2 leakage was 2.3 percent of reheat flow. This will be retested on a monthly basis to determine if the current leakage can be maintained long term using variable clearance packings.

Test results indicate that near design N2 leakage is achievable and can be achieved with standard (i.e. non-variable clearance) packings, however, this level of leakage will not be maintained long term using standard packings. Variable clearance packings can approach design N2 leakage, and theoretically should not degrade as rapidly as standard packings. Since near design leakage can be achieved, this value should be the benchmark for calculating increased operating costs resulting from increased steam leakages.

CONCLUSIONS

The blowdown method has not been successful for measuring N2 leakage on the units we have tested. If the blowdown valve has sufficient flow passing capability to pass all of the N2 leakage flow, this test would be the easiest method to determine the true IP turbine efficiency, however, N2 leakage calculated from a blowdown test is dependent on first stage enthalpy.

The temperature variation method provides a consistent and repeatable method for estimation of N2 leakage and IP efficiency. Proper care must be taken to obtain good unit stability and a reliable and consistent method of obtaining first stage enthalpy must be employed. Note that this method provides an accurate indication of N2 leakage, but IP turbine efficiency is dependent on first stage enthalpy. Temperature variation tests require an upset of normal operations for adequate testing, however the resulting increase in performance test accuracy, and the ability to quantify generation and heat rate losses, is worth the effort.

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