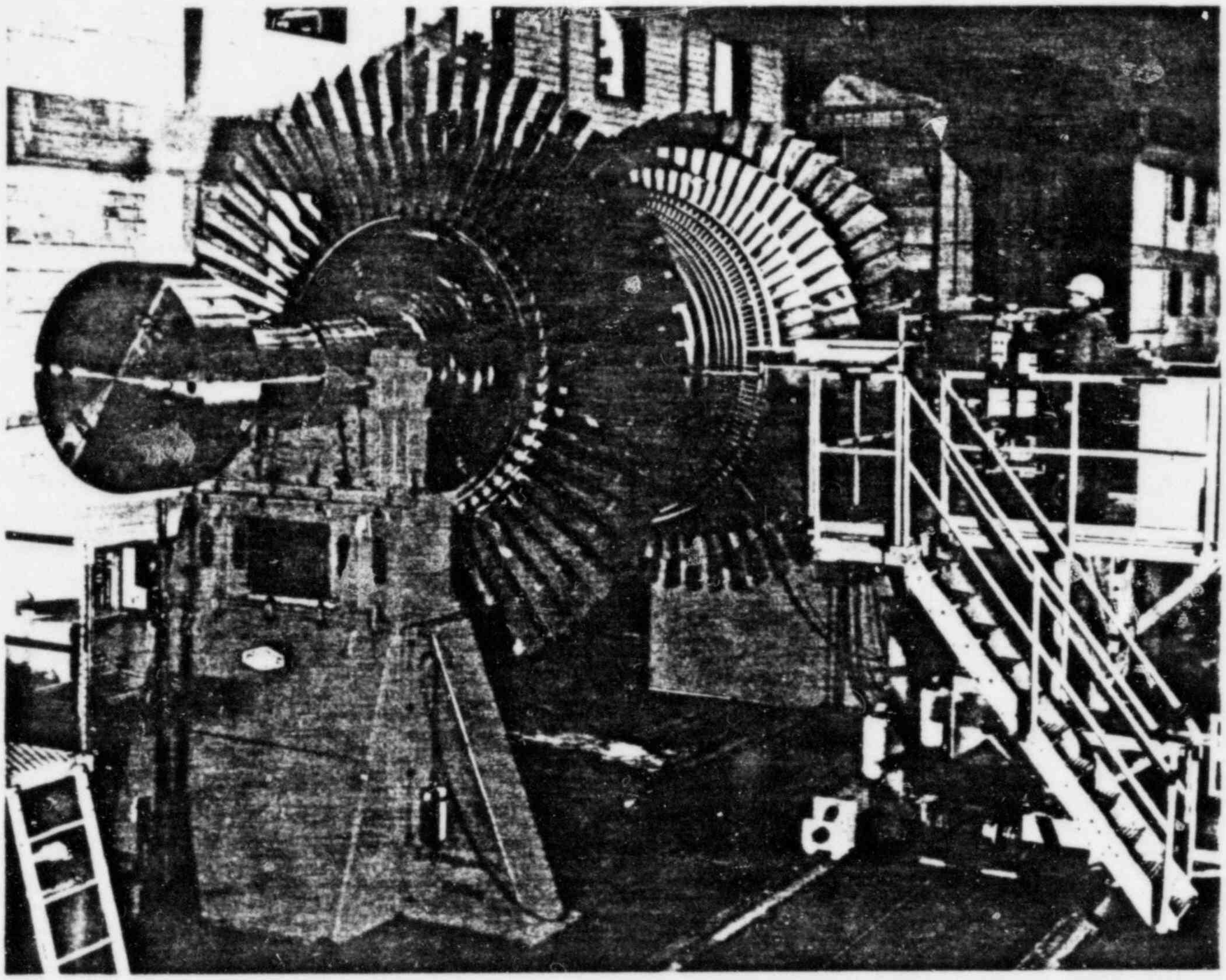
Attachment 2

Design, Operating and Inspection Considerations to Control Stress Corrosion of LP Turbine Disks



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INTRODUCTION

Disk-type LP turbine rotors have been designed for a large variety of power plant applications. The authors' companies and one of their founders, i.e., Siemens, utilize solid forging rotors for all high-speed (3600 rpm and 3000 rpm) turbine-generators. Disk-type rotors have been adopted only for low-speed (1800 rpm and 1500 rpm) turbines in nuclear power plants with PWR and BWR steam supply systems. However, the second founder of Kraftwerk Union, namely, AEG, has supplied disk-type LP turbine rotors for high-speed fossil reheat and non-reheat units.

This paper describes the design, the operating experience and the inspection method of the disk-type LP rotors for nuclear turbine-generators. Additional information about the obsolete AEG disk-type LP turbine rotor design and the operating experience gained in various countries will also be provided.

NUCLEAR LP TURBINE ROTOR DESIGN

The application of solid rotor forgings for high-speed LP turbines is possible because the major forge masters are able to provide high quality forgings of the required size. Independent of the rotor design concept the rough machined weight of even the largest LP rotor is less than 100 metric tons (220,000 lb). Metallurgical inspections and machining of such rotors can be effectively performed. The integrity of these rotors has been proven by excellent operating experience. The solid rotor design concept without axial through-bores reduces the tangential stress level and eliminates corrosion in the rotor center.

Nuclear PWR and BWR steam supply systems provide low-energy steam. As a result, the steam flow of a nuclear turbine is about 70% larger than that of a fossil turbine of equal rating. Also, considering that the output of nuclear units today has risen up to 1300 MW, the application of high-speed turbines cannot be economically justified[1] Low or half-speed LP turbines, developed by the design laws of geometric scaling, feature annulus areas that are four times larger.

The deterrent to building such large turbines was the fact that low-speed LP turbines would require solid rotors weighing 300 metric tons (660,000 lb)[2] Forgings of this size were not available when the first low-speed LP turbines were designed about 25 years ago. Even though such large forgings have recently become available, present nuclear LP turbine rotors are of the disk-type design, which only require a 70 metric tons (154,000 lb) step shaft forging and ten disks with a maximum weight of 10 metric tons (22,000 lb). **Figure 1** shows a low-speed, 4-flow tandem compound nuclear turbine with two disk-type LP turbine rotors.

Developing disk-type rotors required a concerted effort to design a shaft/disk shrink fit with a minimum of tensile stress concentrations in order to provide for the lowest possible susceptibility to corrosive attack.

The regions of greatest concern are the shaft steps, the shrink-fit boundaries and last, but not least, the keyways. Photoelastic model tests, finite element stress calculations and full-size testing in the overspeed facility to measure local stress risers were performed before finalizing the disk/shaft, shrink-fit and keyway configuration shown in **Figure 2**. This design features five cylindrical

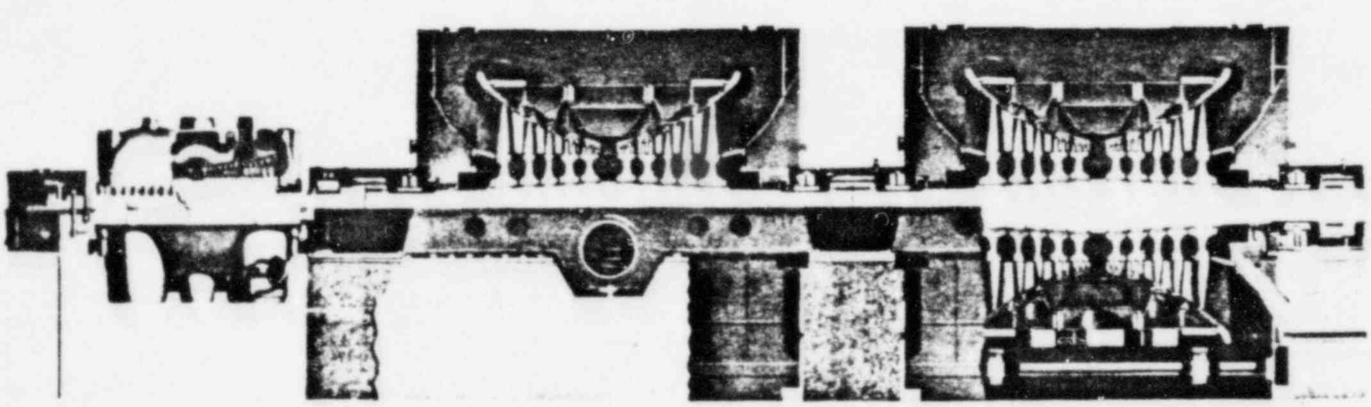

Fig. 1 Low-Speed 1300 MW Tandem-Compound Nuclear Turbine with Disk-Type LP Rotors.

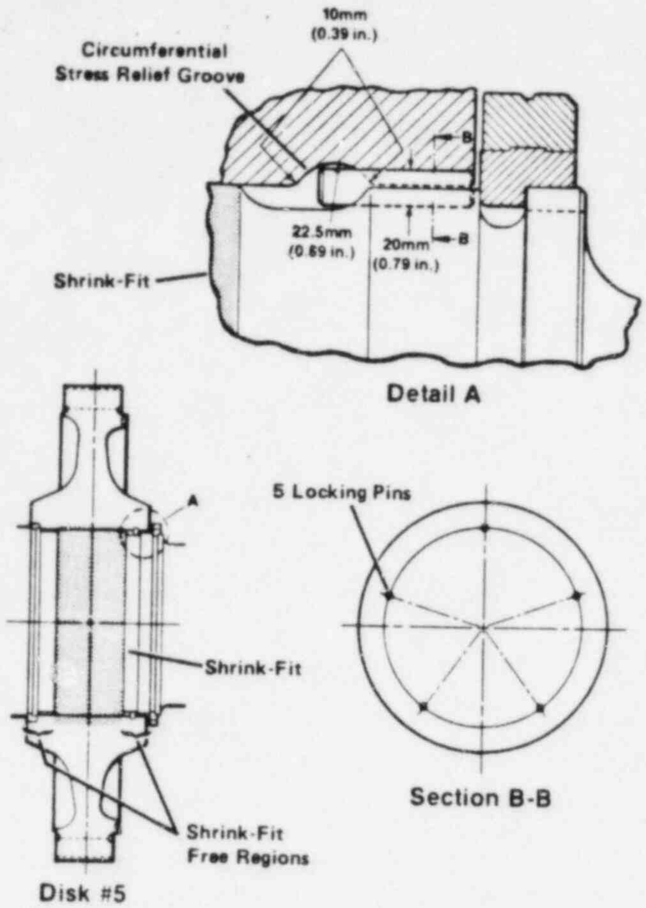

Fig. 2 Shrink-Fit and Keyway Configuration of LP Turbine Disk #5

keys for each disk, assembled into keyways located in the shrink-fit free region on the downstream sides of the disks. The sections adjacent to the shrink-fit areas at each side of the disks, as well as the large radius circumferential stress relief grooves in the shaft and disks are optimized to minimize shrink-fit and keyway stresses and their interaction.

Relating stress corrosion susceptibility of disk keyways only to the evaluated design stresses, however, is inappropriate. Other factors are equally important, such as the formation of crevices and steam/water flow conditions. Design stresses also reflect an unrealistic picture since the factory overspeed test already causes local plastic deformation of the keyway and reduces subsequent local operating stresses as shown by the two examples in **Figure 3**.

Examples 1 and 2 represent two different LP turbine disks with different keyway configurations. The diagram shows the stress-strain curve of the disk material. When shrinking the disks onto the shaft, the keyways of disks 1 and 2 are stressed as indicated by points A_1 and A_2 respectively. Note that the local stressing in keyway 2 is higher than in keyway 1. Stresses increase during the 20% overspeed test as indicated by points B_1 and B_2. However, the stresses in keyway 2 increase by only a small amount because of plastic elongation. Bringing the rotor back to standstill reduces the local stress levels at both keyways to much lower levels as shown by points C_1 and C_2. At this time, the stress level of keyway 2 is lower, but keyway 2 underwent more plastic elongation. Subsequent power plant operation between zero and rated speed results in stressing keyway 1 and 2 between points C_1-D_1 and C_2-D_2 respectively. An overspeed event during plant operation would raise the stresses only to E_1 and E_2, since overspeed is limited to about 10%. A different stress corrosion sensitivity of the two described examples cannot be concluded.

The stress-strain diagram additionally shows how misleading a theoretical evaluation can be when calculating stress levels under elastic material behavior. Such evaluation would not only show much higher design and overspeed stress levels, but also higher operating stresses. Basing the calculations on this wrong assumption would result theoretically in operating stresses roughly 1.1 and 1.5 times higher than yield strength for disks 1 and 2 respectively. (D'_1 and D'_2). However, the stress-strain diagram clearly reveals that under actual material behavior, the operating stresses D_1 and D_2 in disks 1 and 2 are about the same.

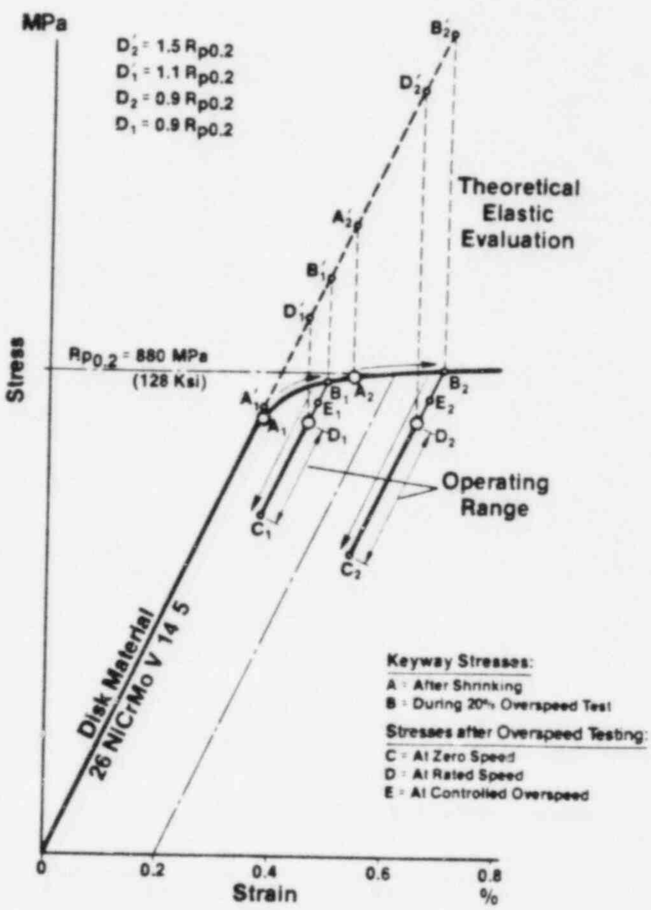

Fig. 3 Stress Levels of Two Different Disk and Keyway Designs.

Longterm experience with various LP disk materials has not revealed significant differences in stress-corrosion behavior. However, extensive testing indicated that disk materials when heat treated to achieve the highest possible yield strength levels (above 1200 MPa or 174 Ksi) became more susceptible to stress corrosion. Nuclear LP turbines of the authors' companies employ disks forged from material 26 NiCrMoV 14 5 similar to ASTM-A471. The disks are modestly heat treated to achieve yield strengths of 1000 MPa (145 Ksi) or less. This modest heat treatment also provides a high fracture toughness. This is important in regard to stress corrosion cracking because a high fracture toughness provides a larger critical crack size, and therefore a longer mean time between in-service inspections.

To date a total of seven million disk service hours have been accumulated with Kraftwerk Union nuclear turbines without a single indication of stress corrosion.

EXPERIENCE WITH DISK-TYPE LP ROTORS OF OBSOLETE DESIGN

Prior to the introduction of the Kraftwerk Union disk-type rotor design for low-speed turbines in 1969, AEG built disk-type LP rotors for fossil and nuclear applications. The disks of these rotors were secured by either rectangular keys or anti-rotation pins as depicted in **Figure 4**. Typically, the first-stage disks employed one rectangular keyway over the entire width of the disk with 2.5 mm (0.1 in.) radii for stress relief. The key was bolted to the rotor with a maximum total side clearance of 0.5 mm (20 mils) and a 0.1 to 0.3 mm (4 to 12 mils) radial clearance between the key and keyway. Each of the remaining disks was secured by three anti-rotation pins located in bores at the disk hub faces. Variations of this design were LP rotors with rectangular keyways in all disks or in only one of the first stage disks. Most of the LP turbine rotors were applied with conical sleeves between the step shaft and the disks.

The non-reheat 200 MW turbine in **Figure 4** with one double-flow LP turbine utilizing eight disks, is a typical example of this obsolete AEG impulse-type turbine design. A large number of such turbines were operating in various countries without any indication of disk cracking, until 1978, when cracks were found in shrunk-on disks of a South African unit [3]

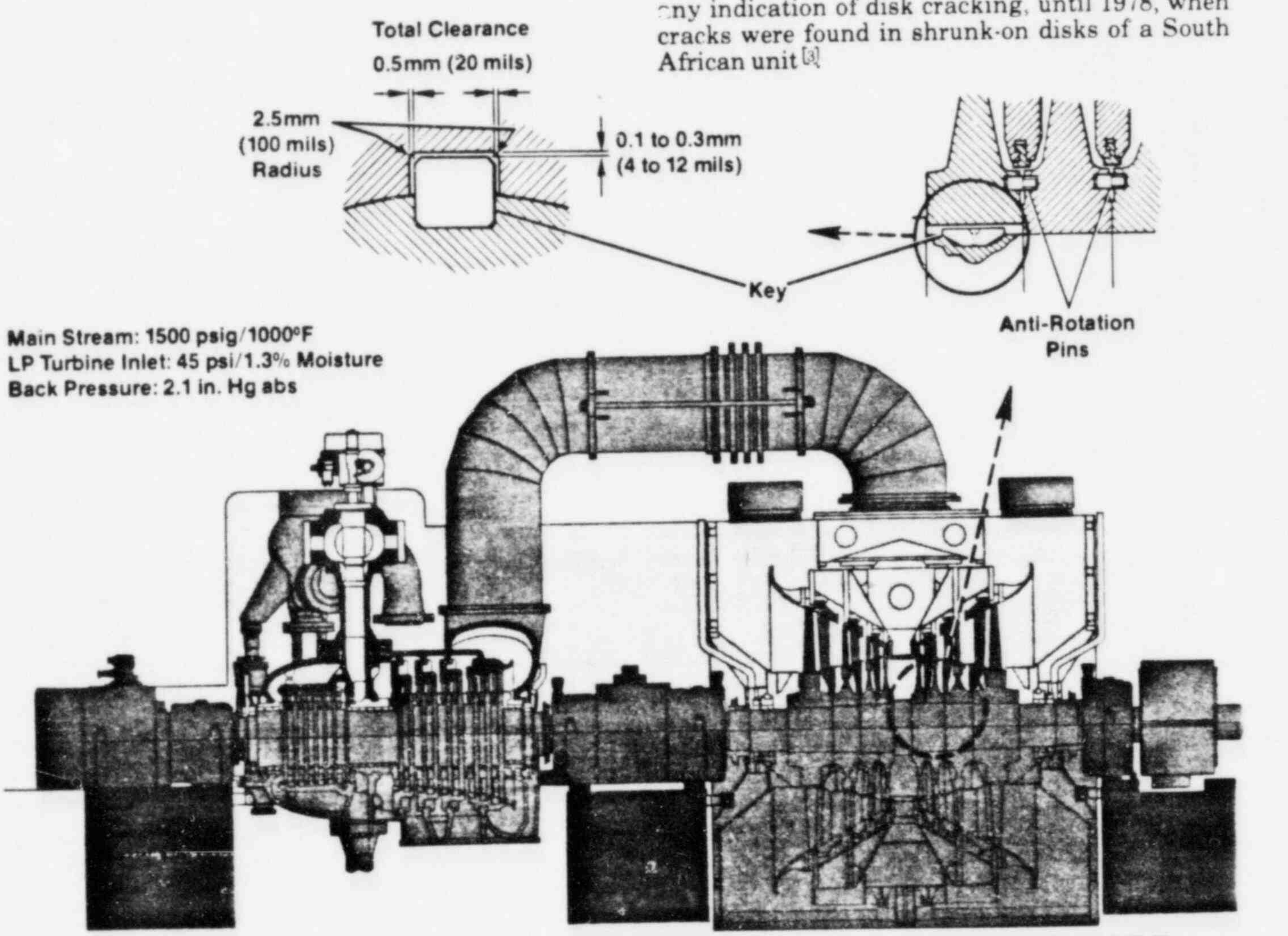

Fig. 4 High-Speed 200 MW Tandem-Compound Non-Reheat Turbine with Disk-Type LP Rotor.

Subsequently, Kraftwerk Union has inspected sixteen of the total of twenty AEG LP rotors in South Africa, after 20,000 to 140,000 service hours. All these turbines are of the non-reheat design, rated between 100 and 200 MW and operate with a small moisture content in the LP turbine admission steam. Disk cracks were found in fifteen of the sixteen turbines in three different power plants. The test results from the 128 disks revealed stress corrosion cracking in 49 disks as listed in **Table I**. In forty disks, cracks were found only in the vicinity of the rectangular keyways. However, seven disks without keyways and two disks with keyways had stress corrosion cracks in the disk hubs, the disk faces and/or the anti-rotation pin holes. Only one disk showed corrosion cracking in the shrink-fit region. All the remaining disk-hub cracks were located in the non-shrink area. The shrink fit area of this disk design covers approximately 60% of the disk width and the adjacent 20% overlaps at each side actually have small clearances. Cracks in areas other than the keyway regions occurred in three of the sixteen turbines after 40,000 to 75,000 service hours.

All the disk cracks found revealed an intergranular crack pattern with branched crack propagation. The crack initiation locations were often pitted and heavily expanded by corrosion. Corrosion products penetrated to the crack tips and were analyzed to be magnetite (Fe_3O_4). None of the cracks had any significant deposits of salts or sodium hydroxide (NaOH). All these findings are typical indications of stress corrosion. The maximum crack depth found in one of the 128 inspected disks was 18 mm or almost 3/4 in. As plotted in **Figure 5**, the inspection results from the three South African power plants indicate a conservative maximum crack propagation rate of 2mm/10^4 hours (approx. 80 mils/10^4 hours) from initial unit start-up.

Other turbine suppliers have discovered similar corrosion cracking in the United Kingdom on fossil-fueled non-reheat turbines. These turbines were inspected after the disk cracking event in the Hinkley Point nuclear power station. Stress corrosion cracking was reported to occur in pure stagnant saturated steam or condensate at elevated temperatures [4]. These temperatures correspond to LP turbine operating temperatures at and below the Wilson line. Stress corrosion cracking of LP turbine disks has also been found in fossil and nuclear power plants in the USA as reported by EPRI [5].

TABLE I
Disk Inspection Results from 16 Non-Reheat Turbines Operating in Three South African Power Plants.

5 LP Rotors with Keyways in all Disks			
	Number of Disks	**Disks with Keyways**	**Disks without Keyways**
Disks Inspected	40	40	0
Corrosion Cracks in Disks	22	22	—
Corrosion Cracks in Keyways only	—	22	—
10 LP Rotors with Keyways only in the First-Stage Disks			
	Number of Disks	**Disks with Keyways**	**Disks without Keyways**
Disks Inspected	80	20	60
Corrosion Cracks in Disks	26	19	7
Corrosion Cracks in Keyways only	—	17	—
1 LP Rotor with Keyway only in one First-Stage Disk			
	Number of Disks	**Disks with Keyways**	**Disks without Keyways**
Disks Inspected	8	1	7
Corrosion Cracks in Disks	1	1	0
Corrosion Cracks in Keyways only	—	1	—
TOTAL RESULT	**Number of Disks**	**Disks with Keyways**	**Disks without Keyways**
Disks Inspected	128	61	67
Corrosion Cracks in Disks	49	42	7
Corrosion Cracks in Keyways only	—	40	—

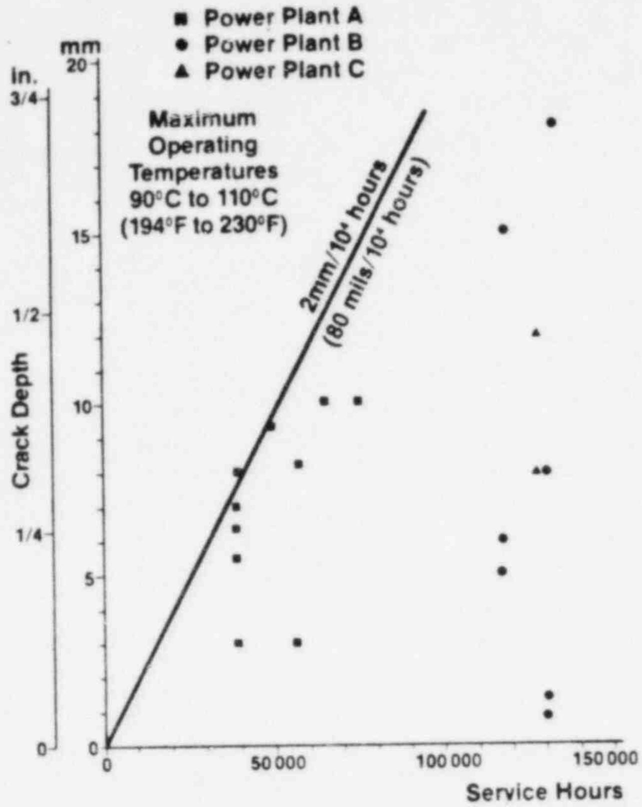

Fig. 5 Keyway Inspection Results of Three South African Power Plants.

After Kraftwerk Union discovered the first corrosion cracks in the South African turbines, an inspection program for all disk-type LP rotors was introduced. In the meantime, several LP rotors have even been disassembled and the disks were tested using magnetic partical techniques. The inspected LP rotors have been operating over long time periods in various European countries, such as Germany, Denmark and Greece. Results of these inspection programs are depicted in **Figure 6**. The disks of seven of these turbines have been inspected and no disk cracking has been discovered after 10 to 20 years of operation.

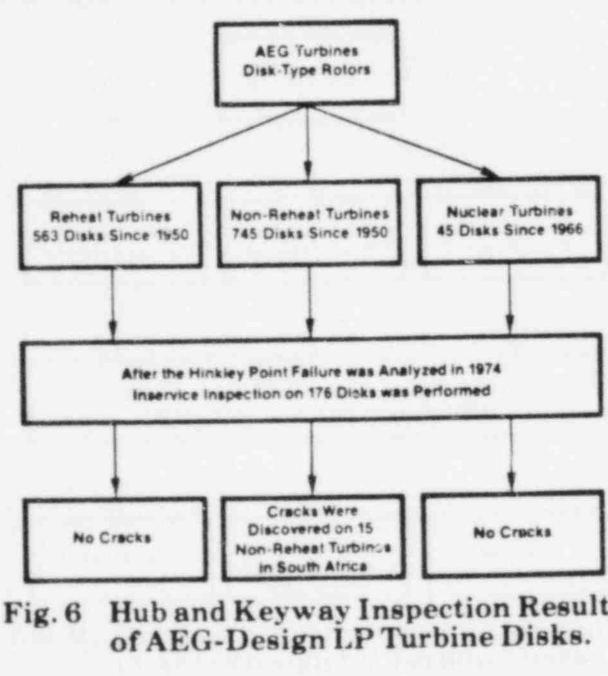

Fig. 6 Hub and Keyway Inspection Results of AEG-Design LP Turbine Disks.

TABLE II
LP Turbine Disk Materials.

MATERIAL	CHEMICAL COMPOSITION IN %				
	Carbon C	Chromium Cr	Molybdenum Mo	Nickel Ni	Vanadium V
28 NiCrMo 7 4 ASTM-A294	0.26/0.32	0.90/1.2	0.30/0.40	1.6/1.8	—
34 CrNiMo 6 ASTM-A294	0.30/0.38	1.4/1.7	0.20/0.30	1.4/1.7	—
26 NiCrMoV 14 5 ASTM-A471	0.26/0.35	1.2/1.7	0.30/0.45	3.4/3.6	<0.15

All these LP turbine disks were forged from low-alloy, heat-treatable steels listed in **Table II**. For most units, the steel 34 CrNiMo 6 (similar to ASTM-A 294) with an $R_{p0.2}$ yield strength of 700 to 1000 MPa (100 to 145 Ksi) after heat treatment was utilized. The heat treating procedure was modified for each specific application to achieve optimal properties. The disks for the last five South African units were forged from 26 NiCrMoV 14 5 steel (similar to ASTM-A 471) with a maximum yield strength of 1000 MPa (approx. 145 Ksi) after heat treatment. A few older LP turbine disks were made from 28 NiCrMo 7 4 steel, which is similar to ASTM-A 294 material.

Tests revealed no significant difference in the susceptibility of these three materials to stress corrosion. Not one of the disks with cracks showed evidence of temper embrittlement. The fracture toughness of the disk material 26 NiCrMoV 14 5 (ASTM-A 471) is at least 178 MPa $\sqrt{m}$ at 20°C (approx. 163 Ksi $\sqrt{in.}$ at 68°F) and the minimum toughness of the older 34 CrNiMo 6 and 28 NiCrMo 7 4 materials is about 95 MPa $\sqrt{m}$ at 20°C (approx. 87 Ksi $\sqrt{in.}$ at 68°F). At 100°C (212°F) however, the fracture toughness of both later materials reaches a higher level which corresponds to the fracture toughness achieved with the presently employed 26 NiCrMoV 14 5 (ASTM-A 471) disk material at 20°C. The high fracture toughness of the three LP turbine disk materials at operating temperature provides a relatively large critical crack size.

POWER PLANT WATER CHEMISTRY

Since the disk inspection results from the South African and the European power plants were so different, a comparison of the power plant water chemistry was conducted. Major differences and typical analysis values from the turbine condensate of European and South African units are shown in **Table III**. The cation conductivity of the turbine condensate is higher in the South African power plants and the pH-level does not correlate with the ammonia content. In general, a high cation conductivity reveals an increased salt content in the condensate, but even under these conditions, the relationship between the pH-level and

TABLE III
Comparison of Power Plant Water Chemistries.

	Europe		South Africa
Boiler Design Make-Up Water Storage Venting of Deaerator Condensate Polishing System	Mainly Once-Through Closed to Atmosphere To Atmosphere Normally 100% Capacity		Drum-Type Open to Atmosphere Into Condenser None
Water Treatment	AVT*	Combined	Coordinated Phosphate
Turbine Condensate Impurities During Normal Operation:			
Conductivity Downstream of a Strong-Acidic Cation Exchanger μS/cm (μmho/cm)	<0.2	<0.2	0.3
Sodium (Na) ppb	<1	<1	<1
Ammonia (NH_3) ppb	600	~50	200-600
pH Level	9.3	~8.5	8.4-9.2
Oxygen (O_2) ppb	<20	200-300	<15
Carbon Dioxide (CO_2) ppb	<100	<100	300-400 calculated

*AVT - All Volatile Treatment

the ammonia content should not diverge. The data from the South African units tend to indicate the presence of free acids in the turbine condensate and consequently, in the steam. Compounds such as carbon dioxide, sulfur dioxide, muriatic acid, or organic acids (formic acid, acetic acid, etc.) could be present[6]. Ongoing investigations indicate that the presence of carbon dioxide CO_2 in the steam is most likely. The logical source for CO_2 would be the make-up water supply, since the make-up water is stored open to the atmosphere. In addition, routing the deaerator vents back into the condenser keeps CO_2 impurities in the system. Because there is also no condensate polishing system, only the degassing in the condensers could actually lower the CO_2 level of the system.

Other possible sources for CO_2 contamination can be the boiler water additives, such as polyphosphate and sodium hydroxide polluted with carbonates. Air in-leakages at the LP turbines, condensers, BFP turbines, LP feedwater heaters, pumps, piping and expansion joints are also sources for CO_2 impurities. Presently, investigations have been started to find the reason for the high cation conductivity of the condensate in the South African power plants.

Even though there are major differences between the European once-through boiler cycles and the South African plant cycles, it should be noted that the South African plants were designed in accordance with the state-of-art for drum-type boiler power stations. Investigations confirmed that all the South African units have been operated with very low salt impurity levels.

In European power plants the cation conductivity is generally smaller than 0.2μS/cm (0.2μmho/cm) which can be attributed to several measures. Make-up water storage tanks are closed to the atmosphere by steam blanketing, nitrogen blanketing or soda lime absorption devices. Additional make-up water is often stored in tanks at elevated temperatures of 105°C (221°F) to 115°C (240°F). Gases from the deaerator are normally vented to the atmosphere. All modern plants are equipped with fullsize condensate polishing systems. All modern boilers are of the once-through design, and are operated without any chemical feedwater treatment other than the injection of hydrazine and ammonia.

LABORATORY TESTING

An extensive stress corrosion investigation program was started in 1978 immediately after Kraftwerk Union found the first corrosion cracks in LP disks of an AEG turbine in South Africa. Smooth and notched test specimens were taken from disks of affected turbines to perform stress corrosion tests up to full 100% yield strength in demineralized water with low oxygen and saturated oxygen content. Test samples from disk material with up to 1000 MPa (145 Ksi) $R_{p0.2}$ yield strength did not crack during the 20,000 hour test period. However, cracking occurred in material samples with higher than 1200 MPa (174 Ksi) yield strength, when stressed to full yield strength. During these tests, the test environment was constantly cleaned by mixed bed ion filters to maintain the low conductivity level of the water.

Figure 7 depicts the results of tests with specimens stressed up to 90% of their yield strength. Adding carbon dioxide to the water cycle resulted in corrosion cracking, even of the low yield strength material after less than 1000 hours. The specimens tested in water saturated with carbon dioxide exhibited extensive corrosion pits and

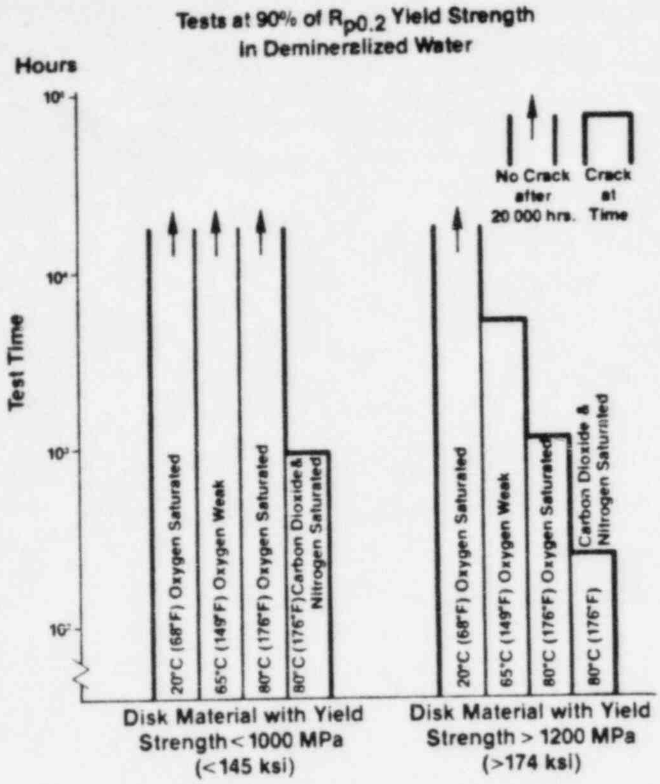

Fig. 7 Stress Corrosion Test Results with Oxygen and Carbon Dioxide.

crack initiation as shown in **Figure 8**. The test sample surfaces were already covered with black oxide (magnetite) shortly after the tests began, whereas samples tested in oxygen poor and saturated demineralized water looked clean and had only a thin, light-colored passive layer even after 20,000 hours of testing. Spot checks with a weak sulfuric acid solution showed results similar to the test in carbon dioxide saturated water.

In conjunction with the tensile tests, compact tension (CT) specimens with fatigue crack initiations, were tested in the laboratory water cycle. Tests in demineralized water showed no indication of crack growth whatsoever. Spot-check type testing must now be confirmed by systematic test programs which have already been started.

With a sufficiently low cation conductivity of the turbine condensate, stress corrosion can practically be eliminated, since low cation conductivity confirms the absence of acidic compounds such as carbon dioxide, sulfur dioxide, muriatic acid, organic acids and possibly salts. Oxygen in the steam or condensate has no influence in this regard. This passivity of oxygen is also confirmed by the fact that BWR and fossil-fueled plants with combined feedwater treatments operate with high oxygen contents. It can also be safely assumed that sufficient alkalinization of the condensate at and below the Wilson line could eliminate stress corrosion or pitting corrosion in LP turbines.

Concentration ratios of ammonia and carbon dioxide in the cycles of the South African power plants, depending on the different distribution

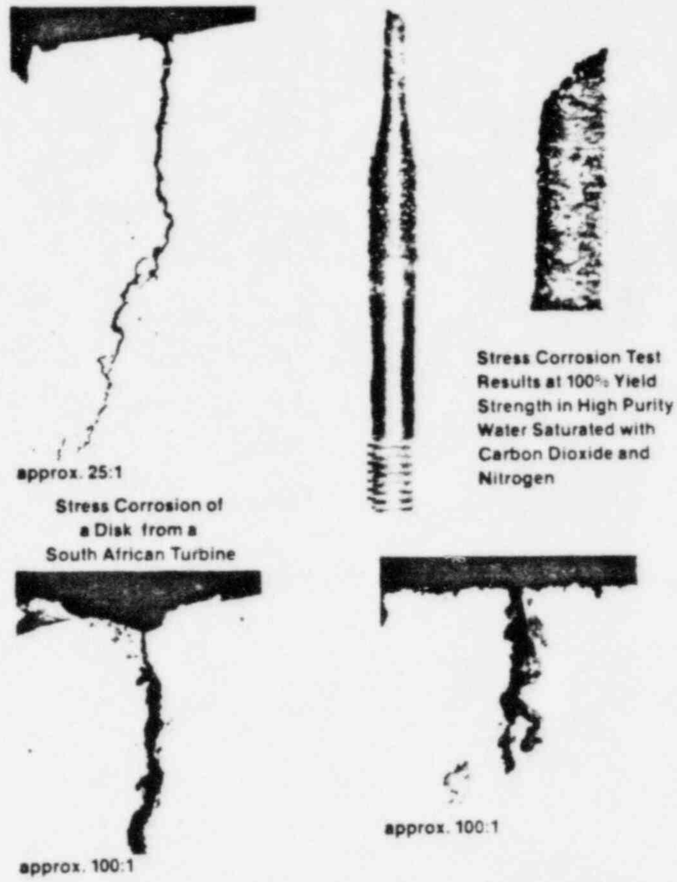

Fig. 8 Stress Corrosion Caused by Carbon Dioxide.

coefficients, most probably cause a significant reduction of the pH-level in the LP turbine at and below the Wilson line. These undesirable conditions do not exist in European power plants because of the absence of carbon dioxide. It is important to note that especially nuclear power plants designed and built by Kraftwerk Union have been carefully scrutinized to eliminate any potential sources for stress corrosion. Routine testing of disk-type LP rotors of these PWR and BWR plants has not revealed any disk cracking. Recently all disks of such an LP turbine were tested both shrunk-on and disassembled from the shaft. The crack-free condition of all the disks and the absence of any corrosive attacks, after 76,000 service hours in a PWR power plant, confirm our laboratory findings.

ULTRASONIC DISK INSPECTION

Inspection of completely assembled and bladed disk-type LP rotors for stress corrosion cracks is a difficult task. The most critical regions of these rotors in regard to stress corrosion are the disk hub bores with their keyways. The ultrasonic inspection technique and test equipment described below has been developed to perform a 100% volumetric hub bore and keyway inspection to detect axial/radial stress corrosion cracks without disassembling the disks from the shaft and without detaching any blades from the disks.

The following conditions had to be considered in developing these in-service inspection methods:

- The ultrasonic probes must be positioned on the disk faces to detect axial/radial cracks.
- The disk face geometries are very complex, complicating the probe positioning for the hub bore and keyway inspection.
- There is only a 50 mm (2 in.) axial distance between the rims of disks 1 and 2 which the probes must clear before they can be positioned on the disk faces.
- The configuration of the keyways complicates the ultrasonic testing and test result evaluation.
- The ultrasonic signals must travel 150 to 650mm (6 to 26 in.) through the disks.
- An acoustic signal transparency with a high coefficient of transmission is to be expected at the shrink fit.

These problems have been solved by optimizing the crack detection capability of the corner reflection method, utilizing a large number of specially designed ultrasonic probes and positioning the probes on the disk faces with the aid of several specially developed mounting plates which are attached to two remote-controlled robot arms.

The corner reflection method provides a double reflection of an ultrasonic signal at a corner formed by the inspected surface and a crack. This reflection signal returns parallel to the sensing signal. In the case of disk inspection, the corner is formed by the disk hub bore and an axial/radial oriented crack.

For the side regions of the disks the pulse/echo technique is applied. An ultrasonic probe, positioned at a specified disk face location, is used as transmitter and receiver. The center regions of the disks are tested with the tandem technique. For this technique probes have to be positioned at each side of the disk. Accurately synchronized positioning with two independent robot arms is required since one probe functions as transmitter and the other one as receiver. (**Fig. 9**).

Prior to the actual tests of shrunk-on disks, investigations were performed to determine the possible acoustic transparency of shrink fits. Perpendicularly applied signals at the shrink fit of a test disk behaved similarly to signals applied to a backwall, indicating that the signal transparency of shrink fits has no or only a minor influence. The next step in preparation for testing was to graphically analyze a complete rotor to define the various probe positions and the number of test regions. This was followed by the determination of required probe characteristics. It was found that a total of 20 to 30 different ultrasonic probe positions on the faces of each disk are needed to perform a 100% volumetric inspection of the disk hub bore with its keyways.

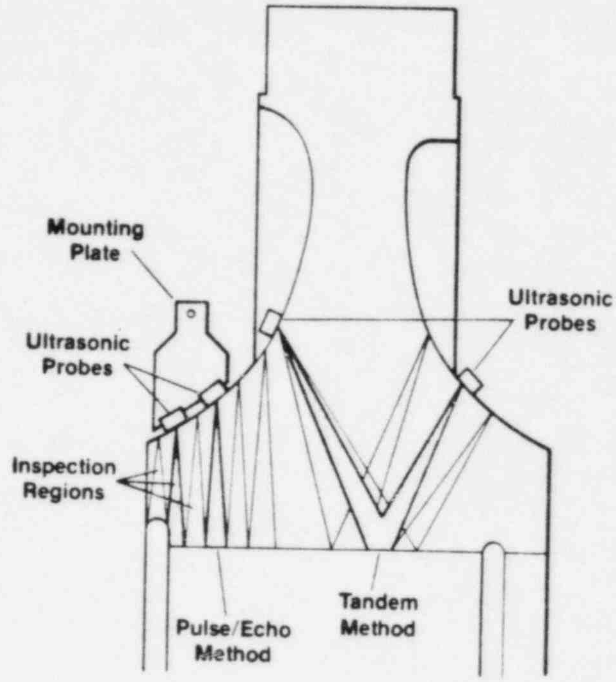

Fig. 9 Ultrasonic Disk Inspection Methods.

The incidence and inclination angles of the probes must then be determined as a function of the probe position at the disk faces and the diameter of the disk hub bore. For a complete LP rotor with two times five disks, a total of two times 80 probe regions with two times 120 probe positions are required, when testing half of the regions with the pulse/echo and the other half with the tandem method. To precisely fulfill the corner reflection criteria for each position, the exact combination of incidence and inclination angle would not be identical for any of the positions, and consequently, 120 ultrasonic probes would be required. However, a somewhat broader range of application can be justified resulting in a reduced number of ultrasonic probes. The investigation revealed that 40 ultrasonic probes in combination with 50 different mounting plates are required to adequately perform the ultrasonic inspection of all ten disks of an LP rotor.

Each mounting plate carries for the pulse/echo test method two ultrasonic probes, whereas for the tandem method only one probe is attached to each mounting plate. Accurate positioning of the mounting plates at the disk faces is controlled by three position transducers. Contact fluid is supplied to each probe and hydraulic pressure is applied to provide for the best possible contact of the probe to the disk face surface (**Fig. 10**).

The test device features two robot arms to simultaneously position two mounting plates at each disk face for the tandem testing of the disk center. Since the axial distance between disk 1 and 2 is only 50 mm (approx. 2 in.), the robot arms have to bring the mounting plates to a vertical position for clearing the disk rims (**Fig. 11**). After passing the

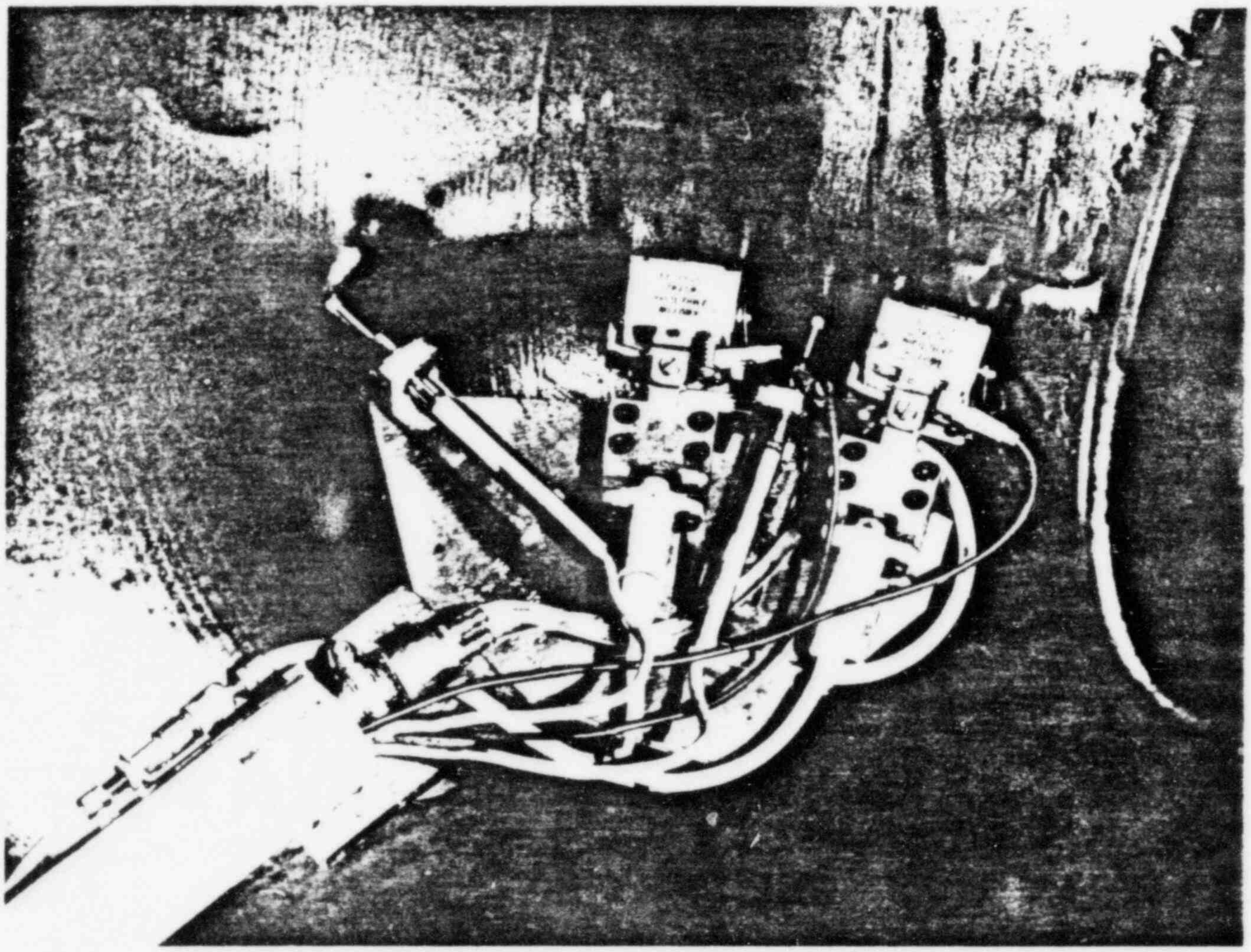

Fig. 10 Ultrasonic Inspection with Two Probes in Position.

rims, the plates are turned 90° into the horizontal operating position. The test probes are then positioned on the disk faces and the ultrasonic inspection commences by turning the rotor at about 1 rpm and testing one specific test region of the hub bore over the entire 360° (**Fig. 12**).

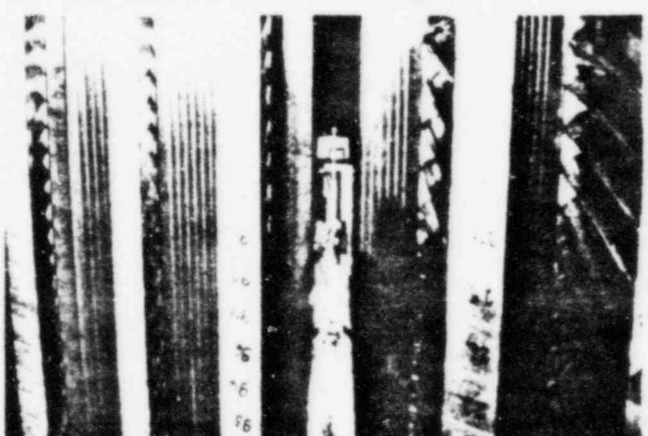

Fig. 11 Mounting Plate in Vertical Position Clearing the Disk Rims.

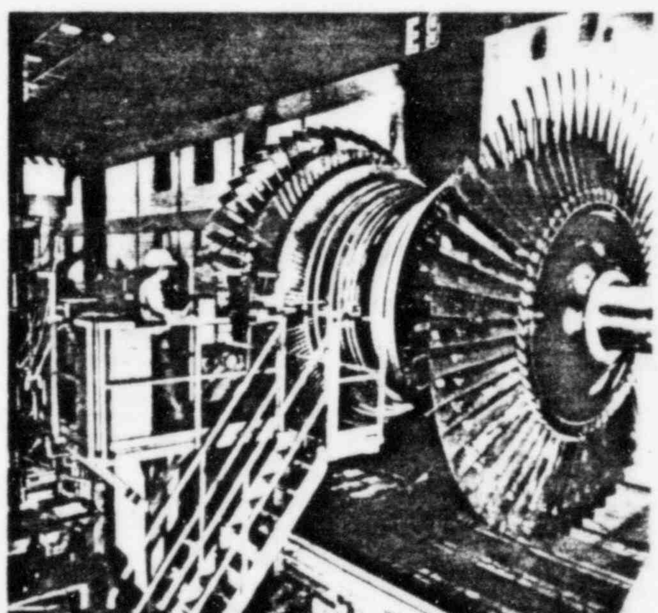

Fig. 12 In-service Inspection of Fully Bladed LP Rotor.

The major components of the test system as depicted in **Figures 12** and **13** are the manipulator with two robot arms to position the ultrasonic probes with the aid of position transducers, the circumferential position measuring device to determine the rotor angle position during testing and the multi-channel ultrasonic test device with line recorder, CRT and magnetic tape data storage.

To check the test technique and inspection system, about 60 axial/radial oriented notches simulating corrosion cracks of different sizes were eroded into the hub bore of a test disk. The disk was shrunk onto a shaft for the purpose of calibrating the inspection system under full-scale conditions. These test results could then be directly related to real disk-type rotor inspections.

This newly developed system was applied for the first time to inspect an LP rotor after roughly 76,000 service hours in a German PWR nuclear power plant. The 100% volumetric inspection of the hub bores and keyways was performed on a completely bladed LP rotor. The ultrasonic testing showed a maximum indication of a 3.3 mm (0.13 in.) equivalent flat bottom hole over a maximum axial distance of 100 mm (4 in.) in one of the ten disks at the shrink fit.

As part of the development program of this specific LP rotor test all disks were removed from the shaft after the ultrasonic inspection. The ultrasonic indication on the aforementioned disk was found to be caused by score marks with a maximum depth of 0.5 mm (20 mils). The score mark configuration clearly indicated that scoring had occurred during the shrink-on and not the shrink-off process (**Fig. 14**). All the shrunk-off disks were closely inspected without indication of any stress corrosion after approx. 76,000 hours of operation.

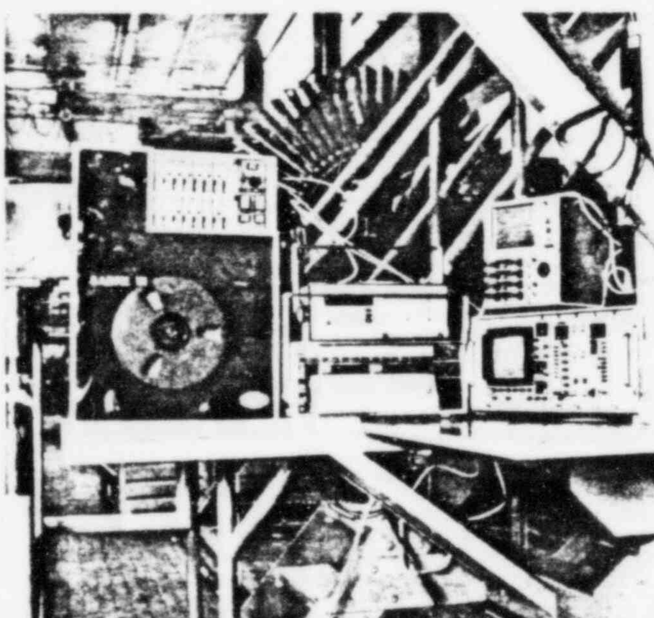

Fig. 13 Ultrasonic Test Equipment.

Fig. 14 Installation Score Marks at the Shrink Fit of a Disk.

CONCLUSION

While minor design modifications to upgrade the resistance of LP turbine disks to stress corrosion may still be possible, the actual improvement from methods such as different anti-rotation devices, sealing of the hub region, coating of the shrink fit or adding keyway ventilation is questionable. Only long-term experience would allow drawing any conclusions about such methods.

The step toward the application of solid rotors would most certainly provide a new concept with improved and predictable corrosion performance. Stress corrosion in the hub section would be completely eliminated with this "mono-bloc" rotor design. It must, however, be noted that such an LP rotor design requires a 300 metric tons (660,000 lb) rotor forging. Such a forging must be machined to about the same configuration as the disk-type rotors presently applied, because much heavier non-contoured rotors would require much larger LP turbine components such as bearings and support structures, which are not presently considered in LP turbine design concepts. Also, non-contoured rotors would not be interchangeable with present disk-type rotors and could, depending on the missile analysis concept, increase the mass and energy of hypothetical missiles.

Most important in controlling stress corrosion is the water/steam cycle chemistry. Units operated without any major cooling water and air in-leakages have shown no stress corrosion problems. It is necessary to avoid not only cooling water in-leakage, but also air in-leakages of low-pressure plant components. Air in-take with the make-up water must also be eliminated. Large condensate polishing systems minimize condensate impurities when operated properly. Sufficient degassing of the feedwater cycle is another important method to keep impurities away from the steam generator and the turbine.

In plants where stress corrosion attack is suspected, testing of LP turbine disk-type rotors is recommended after 5 to 10 years operation. The ultrasonic inspection method developed by Kraftwerk Union, allows the detection of axial/radial stress corrosion cracks larger than 10 mm (0.4 in.) in depth. This sensitive 100% volumetric inspection method provides a large mean time between in-service inspections. For prolonging the period of time between early in-service inspections, a benchmark inspection may be justified before shipping. An inspection method is presently under development to precisely determine crack size and growth. It is our goal to develop a system which can measure stress corrosion crack depths in the entire hub bore and keyway region of disks with less than 5 mm (0.2 in.) tolerance.

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[3] K. Schleithoff, "Stress Corrosion Cracking on Steam Turbine Components — Case Histories, Laboratory Tests and Service Experience", Paper presented at EPRI/CERL Workshop on Turbine Disc Cracking, Central Electricity Research Laboratories, Leatherhead, England, November 28-30, 1979.

[4] J.M. Hodge and I.L. Mogford, "UK Experience of Stress Corrosion Cracking in Steam Turbine Discs", Proceedings, The Institution of Mechanical Engineers, Vol. 193, No. 11, 1979, pp. 93-109.

[5] EPRI Report NP-2429-LD, Vol. 1-7, "Steam Turbine Disc Cracking Experience, Research Project 1398-5, Final Report", June 1982.

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ATTACHMENT 3

European KWU Nuclear Turbines

(Low Speed-1500 rpm Units)

Unit	MW	Commercial Operation
Stade	662	1976
Biblis A	1200	1975
Biblis B	1300	1977
Wuergassen	670	1975
Brunsbuettel	806	1977
Unterweser	1300	1979
Isar	907	1979
Philippsburg	864	1980

ATTACHMENT 3

Utility Power Corporation

Return TO REK

R. J. GARY

EXECUTIVE VICE PRESIDENT

TEXAS UTILITIES GENERATING CO.

Utility Power Corporation
2615 LBJ Freeway, P.O. Box 344109, Dallas, Texas 75234 / (214) 247-4511

November 13, 1981

Mr. R. J. Gary
Executive Vice President
Texas Utilities Generating Company
2001 Bryan Tower
Dallas, Texas 75201

Dear Bob:

In response to your request, enclosed is a listing of the operating statistics as well as explanations for each plant outage for those large nuclear power plants in Europe with Kraftwerk Union turbine generators. The data presented covers operation for the years 1976 through 1980.

As you recall, the 54" last stage blade running at 1500 rpm is in the same blade family as the 44" last stage blade for Comanche Peak and Grand Gulf. The 22" last stage blades on both Handley units are also of the same blade family.

If additional information is needed, please do not hesitate to contact us.

Very truly yours,

Gary M Cook

Gary M. Cook
Regional Vice President

Enclosure

GMC:pm

STADE 662 MW, TC-4F-54, 1500 RPM

Commercial Operation : May 19, 1972

Operation Statistics From January 1, 1976

To December 31, 1980

	1976	1977	1978	1979	1980
Power Generation, MWHRS	5,452.3	5,430.5	5,518.2	4,637.0	4,347.3
Availability Factor, %	94.5	94.1	95.1	77.3	76.7
Cum. Availability Factor, %	84.5	86.2	87.7	86.3	85.2
Capacity Factor, %	93.9	93.6	95.4	76.5	76.2
Cum. Capacity Factor, %	83.3	85.1	86.7	85.4	84.3

LIST OF OUTAGES BY YEAR

Cause and Duration

1976

1) Turbine extraction leaks (9 hrs)
2) Fourth refueling outage (455 hrs)
3) Reactor scram during reactor protection test and leak in secondary cooling loop valve (7.5 hrs)
4) Outage of main reactor coolant pump No. 3 (0.5 hrs)
5) Turbine trip caused by lightning striking outdoor switching station (8 hrs)
6) LP preheater repair (5 hrs)

1977

7) 5th refueling outage scheduled maintenance inspection work (473 hrs)
8) Voltage collapse due to lightning striking the 220 kV power line causing outage outage of main reactor coolant pumps Nos. 3 and 4 (1.5 hrs)
9) Repair of the HP turbine exhaust end flange (6 hrs)
10) Repair broken wire in the control voltage supply of the feedwater pumps (1 hr)
11) Repair of the drain piping for the HP turbine extraction (9 hrs)
12) Leak in the closed cooling water piping (3 hrs)

1978

13) Bypass pipe was drilled during field service work (1 hr)

14) Unscheduled shutdown because of defective measuring device (0.7 hrs)

15) 6th refueling outage (403 hrs)

16) Scheduled shutdown: turbine overspeed test (1 hr)

17) Unscheduled shutdown because of leak at check valve in main steam pipe and condenser tube rupture (6 hrs)

18) Leaky seal in pressure relief drain system (6 hrs)

19) Reactor trip (2 hrs)

20) Leak in secondary loop (7 hrs)

1979

21) 7th refueling outage (1924 hrs)

22) Functional test of main steam valves and overspeed test (12 hrs)

23) Reactor trip during test of reactor safety protection (1 hr)

24) Leaky condensate drain pipe on feedwater tank (47 hrs)

1980

25) 8th refueling and maintenance outage (2065 hrs)

26) Test of overspeed and main steam valves (11 hrs)

27) Reactor trip due to defective level measurement in steam-generator No. 4 (2 hrs)

BORSSELE, 469 MW, TC-6F-35, 3000 RPM

Commercial Operation : October 26, 1973

Operation Statistics From January 1, 1976

To December 31, 1980

	1976	1977	1978	1979	1980
Power Generation, MWHRS	3,444.8	3,329.3	3,628.9	3,083.1	3,706.9
Availability Factor, %	85.5	83.2	91.3	77.5	96.7
Cum. Availability Factor, %	78.8	79.6	81.7	81.0	83.3
Capacity Factor, %	83.9	80.6	88.4	75.2	92.3
Cum Capacity Factor, %	73.4	75.4	77.8	77.4	79.9

LIST OF OUTAGES BY YEAR

Cause and Duration

1976

1) Feed pump outage (4 hrs)
2) Second refueling (1236 hrs)
3) Secondary drain valve leaks (10 hrs)
4) Leak in hydraulic accumulator pump (19 hrs)

1977

5) 3rd refueling and scheduled maintenance inspection work (1128 hrs)
6) Actuation of the vacuum safety valves (4 hrs)
7) Refueling and maintenance inspection (413 hrs)

1978

8) Unscheduled shutdown because of malfunction in the reheater circuit during changeover; valve was closed rather than opened (4 hrs)
9) Unscheduled shutdown because of outage of main feed pumps (9 hrs)
10) Unscheduled shutdown because of oil leak at submersible pump (8.5 hrs)
11) Turbine trip during checkout of automatic turbine protective system (2 hrs)
12) 5th refueling (768 hrs)
13) Unscheduled shutdown for balancing of LP rotor (6 hrs)
14) Unscheduled shutdown for balancing of LP rotor (7 hrs)

1979

15) Leak at main steam pipe of HP turbine (252 hrs)

16) Installation of insulation in hot reheat system (LP bypass) (4 hrs)

17) Power network failure. Outage of 150 kV transformer (9 hrs)

18) Leak at compensator of HP turbine (257 hrs)

19) 6th refueling and maintenance inspection (1223 hrs)

20) Malfunction of load control, turbine (7 hrs)

21) Outage of feedwater pumps (12 hrs)

1980

22) Malfunction of generator voltage regulation (3 hrs)

23) Fire at auxiliary station transformer (96 hrs)

24) Replacement of insulator on generator transformer (50 hrs)

25) Reactor Trip via activity measuring point (3 hrs)

26) Repair of main coolant pump seal (137 hrs)

27) Outage of oil supply for main coolant pump (1 hr)

BIBLIS A 1200 MW, TC-64-54, 1500 RPM

Commercial Operation : February 25, 1975

Operational Statistics From January 1, 1976

Through December 31, 1980

..	1976	1977	1978	1979	1980
Power Generation, MWHRS	5,436.9	6,567.7	7,524.3	7,027.4	4,107.4
Availability Factor, %	52.6	67.3	74.6	85.7	46.9
Cum. Availability Factor, %	68.6	68.2	70.0	73.2	68.4
Capacity Factor, %	51.3	62.9	75.2	85.6	43.6
Cum. Capacity Factor, %	68.4	66.4	68.0	71.5	66.2

LIST OF OUTAGES BY YEAR

Cause and Duration

1976

1) Reactor protection system shutdown (126 hrs)
2) Steam leakage at inlet chamber of HP turbine (51 hrs)
3) Malfunction of generator load controller (8 hrs)
4) Faulty actuation of "generator short circuit to ground" alarm (2 hrs)
5) Malfunction of secondary loop control (5.5 hrs)
6) Generator H_2 cooler leakages (37.5 hrs)
7) Condensate system leak (3 hrs)
8) 1st refueling and scheduled maintenance work: (3377 hrs)
   - repairs on main coolant pumps for improving bolted joints
   - extensive tests and repairs on feedwater tank
9) Test of feedwater tank (656 hrs)

1977

10) Physical (radiological) tests imposed by the regulatory authorities (202 hrs)
11) Repair of HP turbine shrouds (299 hrs)
12) Repair of leaky pressurizer spray valves (72 hrs)
13) Replacement of main coolant pump seal (96 hrs)
14) Replacement of main coolant pump frame (72 hrs)
15) Checking of bolt locking devices of main coolant pump motors (21 hrs)
16) Outage of main coolant pump seal water (2 hrs)

17) Malfunction of main steam pressure controller (3 hrs)

18) Replacement of turbine load controller (6 hrs)

19) Sealing of LP preheater pipes (11 hrs)

20) Malfunction of turbine speed measuring unit (7 hrs)

21) Vacuum reduction in the condensers (1 hrs)

22) Broken valve spindle (6 hrs)

23) Scheduled repair of main steam inlet piping (5 hrs)

+24) Scheduled shutdown for replacement of generator primary water filter (4 hrs)

25) Maintenance inspection (1,317 hrs)

26) H_2 leak in the generator (54 hrs)

27) Expansion joint repair in suction line of one main feedwater pump (53 hrs)

28) Repair of emergency diesel (24 hrs)

1978

29) Turbine trip during exchange of an electronic circuit card in the turbine load control system (1 hr)

30) Unscheduled turbine trip because of alarm signal LP preheater level too high, the same occurred during return to service with HP preheater (1 hr)

31) Scheduled short-term shutdown: elimination of leakage in secondary reactor loop (3 hrs

32) Turbine trip following reactor trip: leakage in HP preheater branch 2 (1.5 hrs)

33) Scheduled shutdown: elimination of leaks in reheater condensate cooler and HP preheater (24.5 hrs)

34) Turbine trip and reactor trip occurred following a switching operation in the transformer unit (8 hrs)

35) Turbine trip and reactor trip resulted from a rotor short to ground in the generator (214 hrs)

36) Malfunction of generator circuit breaker due to high humidity (66 hrs)

37) 3rd refueling (2176 hrs)

1979

38) Retrofit of thermocouples in loop lines 2, 3 and 4 (1 hr)

39) Faulty actuation, NZ 16 main steam activity measuring point (2 hrs)

40) Shutdown for inspection of reactor pressure vessel because of structure-borne ultrasonic indication (780 hrs)

41) During repair of coolant pressure control system, Reactor tripped (6 hrs)

42) Repair of secondary drain pipe (6 hrs)

43) Repair of A6 extraction (10 hrs)

44) Malfunction of speed monitor (reference limiter) Turbine Trip (52 hrs)

45) Examination and remounting of emergency backup line between units A and B (92 hrs)

46) Repair of bracket for turbine control valve No. 3 and sealing of HP preheater (16 hrs)

1980

47) Alarm signal, HP preheater level too high (1 hr)

48) 4th refueling and maintenance inspection (20 hrs)

49) Shutdown extension since compact (waste) storage facilities were not available (24 hrs)

50) Main coolant pump outage (oil supply for motor) (3 hrs)

51) Repair of main coolant pump motor (109 hrs)

52) Leak at shut-off gate valve (46 hrs)

53) Leak in recuperative heat exchanger (31 hrs)

54) Alarm signal, coolant pressure controller (2 hrs)

WUERGASSEN, 670 MW, TC-4F-59, 1500 RPM

Commercial Operation : November 11, 1975

Operation Statistics From January 1, 1976

To December 31, 1980

	1976	1977	1978	1979	1980
Power Generation MWHRS	3,840.7	3,793.5	2,857.7	1,598.9	3,969.3
Availability Factor, %	82.8	81.1	62.3	35.8	94.8
Cum. Availability Factor, %	83.5	82.5	76.0	66.3	71.8
Capacity Factor*, %	66.1	64.6	48.7	27.2	67.4
Cum. Capacity Factor*, %	65.3	65.2	60.1	52.2	55.1

* includes 20% mandatory load reduction to 540 MW

LIST OF OUTAGES BY YEAR

Cause and Duration

1976

1) Elimination of a leak on main steam instrument line (12.5 hrs)
2) Elimination of a turbine leak, in the steam pipe between HP and IP section (70.5 hrs)
3) Elimination of leak on check valve in feed water line (22.5 hrs)
4) Elimination of H_2 leaks on generator (57 hrs)
5) Malfunction of pressure regulator (4 hrs)
6) Damage to thrust bearing of turbine - speed governor outage (129.5 hrs)
7) Scram after error made during isolation of a sub-distributor (46.5 hrs)
8) Scheduled shutdown for refueling (1060.5 hrs)
9) Elimination of valve leaks in pressure suppression system (65 hrs)

1977

10) Elimination of auxiliary equipment leaks in the pressure suppression system (31 hrs)
11) Trip because of malfunction of the initial pressure regulation (5 hrs)
12) Defective fuse in the generator exciter unit (19 hrs)
13) Shutdown because of malfunction in the waste gas unit (23 hrs)
14) 3rd refueling and annual maintenance inspection (1417 hrs)
15) Unscheduled shutdown because the emergency diesels did not jointly run up to speed (82.5 hrs)
16) Shoutdown for steam dryer inspection (93 hrs)
17) Unscheduled shutdown because of high neutron flux (8 hrs)

1978

18) Unscheduled shutdown because of repair of (auxiliary) seal steam supply valve (34 hrs)

19) Unscheduled shutdown because of interchange and replacement of auxiliary seal steam supply valves (34 hrs)

20) Unscheduled shutdown because of malfunction in the waste gas unit (10 hrs)

21) Unscheduled shutdown because of leaky valve gland seal in the turbine building (1 hr)

22) Scheduled standstill for inspection of the steam drier. Cracks were found in the partition plates (2101 hrs)

23) Unscheduled shutdown because of malfunction in the compressed air supply for the (suxiliary) seal steam supply valve and additional load (11 hrs)

24) Unscheduled shutdown because of fractured blade in the LP section of the turbine. In addition, a weld joint in the main steam pipe was checked and repaired because of I.D.-O.D. mismatch. (1110 hrs)

25) Arranged shutdown because of malfunction in the control air supply (5 hrs)

26) Unscheduled shutdown because of defective main steam instrument line in the pressure suppression system (44 hrs)

27) Arranged shutdown because of repair to the auxiliary seal steam supply valve (5 hrs)

1979

28) Instrument line for humidity measurement torn off turbine (10 hrs)

29) Instrument line for humidity measurement torn off turbine (7 hrs)

30) Blocking of auxiliary seal steam supply valve. Cause: solenoid coil burned out (67 h-

31) Defective timing element in functional group "waste gas" (6 hrs)

32) Defective timing element in functional group "waste gas" (8 hrs)

33) 4th refueling and annual maintenance inspection, inspection and repair of various weld joints (5256 hrs)

34) Assembly error on a jet pump head in reactor pressure vessel (169 hrs)

35) Short circuit in turbine initial pressure regulator because of incorrectly soldered joint (5 hrs)

36) Malfunction in hydraulic system for isolation valves (4 hrs)

37) Mechanical malfunction of suction gate valve in recirculation loop (51 hrs)

38) Leaky capped stub on relief pipe (19 hrs)

39) Improperly adjusted limit switch on valve connecting line main steam emergency condenser system (5 hrs)

1980

40) Mechanical malfunction of recirculation loop valve (4 hrs)

41) Leaky instrument line for main steam flow rate measurement (34 hrs)

42) Inaccurate setting at recirculation loop valve (2 hrs)

43) Leak in pressure suppression system because of crack in an instrument line for main steam measurement (60 hrs)

44) Leak in pressure suppression system because of crack in an instrument line for main steam measurement (21 hrs)

45) Oil fire at turbine resulting from oil leak at a control line (21 hrs)

46) Leak at turbine drain (2 hrs)

47) Outage of No. 2 control air compressor for relief system (40 hrs)

48) Relief valve test, limited movement of solenoids for pre-control valves because of hardened rubber sleeves (61 hrs)

49) 2 functionally inoperative isolation valves due to mechanical damage of stuffing box (222 hrs)

BIBLIS B, 1300 MW, TC-6F-54, 1500 RPM

Commercial Operation : January 31, 1977

Operational Statistics From February 1, 1977

To December 31, 1980

..	1977	1978	1979	1980
Power Generation, MWHRS	7,802.9	6,164.3	6,517.4	5,984.4
Availability Factor, %	86.6	68.7	82.8	65.6
Cum. Availability Factor, %	86.6	77.0	79.0	75.4
Capacity Factor, %	75.2	63.6	81.7	61.8
Cum. Capacity Factor, %	75.2	69.1	73.4	70.9

LIST OF OUTAGES BY YEAR

Cause and Duration

1977

1) Outage of control oil supply, generator-end (21 hrs)
2) Scheduled inspection of No. 3 LP bearing (10 hrs)
3) Remeasuring & localizing of rotor short circuit to ground (22 hrs)
4) Generator maintenance inspection (768 hrs)
5) Intentional manual coastdown for elimination of leakage at the main steam inlet pipe & at the HP casing (10 hrs)
6) Turbine trip actuation because of faulty triggering of signal, generator main current lead because of bypass system outage: reactor trip (5 hrs)
7) Cooling water outage in transformer closed cooling loop (4 hrs)
8) Rotor ground fault at generator stator and leak elimination in secondary system (236 hr
9) Changing of an electronic circuit card in rod control system led to actuation of reactor trip via power limiter (5 hrs)
10) During startup: outage of a main feed pump resulting in turbine trip actuation (3 hrs)

1978

11) Shutdown for elimination of an H_2 leak in the generator (52 hrs)
12) Main coolant pump seal replaced and reheater repaired (509 hrs)
13) Unscheduled shutdown for elimination of a leak on the condensate collecting tank (8 hrs
14) Physical (radiological) tests specified by the regulatory authorities (25 hrs)

15) Unscheduled shutdown because of crack in a weld joint behind the seal-steam condenser (seal steam system) area (13 hrs)

16) 1st refueling (1734 hrs)

17) Manual shutdown - cracks in the dryer connection on the main steam inlet pipe of the HP turbine (52 hrs)

18) Turbine trip and reactor trip — electrical defect in the automatic turbine tester (2 hrs)

19) Replacement of main coolant pump seal (333 hrs)

20) Faulty signal from measuring point (4 hrs)

21) Unscheduled shutdown for cleaning of a filter in the generator cooling circuit (5 hrs)

1979

22) Rotor ground fault (leakage) resistance too low (31 hrs)

23) Filter replacement in generator cooling circuit (6 hrs)

24) Checking of filters in generator cooling circuit (4 hrs)

25) False actuation Phase R Turbine Trip (1 hr)

26) Repair of leak in turbine drain (weld joint) (3 hrs)

27) Defective adapter plugged in (outage of main feedwater, reactor trip) (1 hr)

28) Momentary load increase of reactor after faulty lowering of 4 control rods Reactor Trip (7 hrs)

29) Alarm "stator winding flow rate too low", turbine trip, cause unclear (1 hr)

30) 2nd refueling (1325 hrs)

31) Examination and remounting of emergency backup line between units A & B (92 hrs)

32) Repair of bracket for turbine control valve No. 3 (9 hrs)

1980

33) Leakage at the generator circuit breaker (22 hrs)

34) Faulty actuation of signal for feedwater tank, level too low, Reactor Trip (2 hrs)

35) Repair of LP pre-heater (18 hrs)

36) Malfunction in control of main stop valves (4 hrs)

37) Faulty actuation, generator stator cooling control (1 hr)

38) 3rd refueling and maintenance inspection, extension of the shutdown period due to repairs of core baffle bolts and main steam pipes Loop 2 & 3 (2800 hrs)

BRUNSBUETTEL, 806 MW, TC-4F-54, 1500 RPM

Commercial Operation : September 2, 1977

Operational Statistics From September 2, 1977

To December 31, 1980

	1977	1978	1979	1980
Power Generation, MWHRS	3,466.0	2,437.9	0	750.9
Availability Factor, %	51.7	38.9	0	32.2
Cum. Availability Factor, %	49.8	44.0	28.8	29.7
Service Factor, %	49.9	34.7	0	27.4
Cum. Service Factor, %	47.7	45.1	26.7	26.9

LIST OF OUTAGES BY YEAR

Cause and Duration

1977

1) Reactor trip (4 hrs)
2) Generator protection trip as a result of underexcitation (11 hrs)
3) Reactor trip. Outage of steam jet air ejector (5 hrs)
4) Shutdown of unit for repair of a valve leak in secondary condensate circuit (30 hrs)
5) Reactor scram due to drifting of a level measuring line (66 hrs)
6) Repair of valve leaks (41 hrs)
7) Feedpump outage (13 hrs)
8) Faulty closing of turbine trip valve (8 hrs)
9) Outage of one condensate pump (4 hrs)
10) Shutdown because of faulty closing of isolation valve (1 hr)
11) Scheduled shutdown for checking reactor protection system with simultaneous maintenance inspection of various unit components (703 hrs)
12) Unscheduled shutdown for repair of valve leaks (44 hrs)
13) Unscheduled shutdown for repair of No. 2 axial pump and shutdown of No. 3 axial pump as well as work in the turbine condenser area (98 hrs)
14) Unscheduled shutdown for repair of No. 3 axial pump (3213 hrs)

1978

15) Jammed control rod and replacement of the rear turbine bearing in the HP section (186 hrs

16) Reactor trip during repeat test of inverter for reactor protection (4 hrs)

17) Leaky gland seals on three valves of (pressurized) bearing water system and because of a gland seal leak-off pipe which was not fully penetration welded, as well as a leaky gland seal on a valve of the (pressurized) bearing water system within the pressure suppression system (146 hrs)

18) Malfunction in the feed pump area (10 hrs)

19) Turbine trip because of fluctuation in 380 KV power network during changeover on load distributer (1.5 hrs)

20) Erroneous measurement of generator cooling water temperature (5 hrs)

21) Shutdown during changeover of scram accumulator tank (6 hrs)

22) Shutdown for inspection (4704 hrs)

1979

22) Continued - Outage, No commercial operation due to inspection work (8760 hrs)

1980

22) Continued - Shutdown because of malfunction on 6/18/78. Finishing of repair work and improvement of unit in accordance with requirements of the German Ministry of the Interic the licensing authority, the Reactor Safety Commission, the government examiner, and the Supreme Court decision of 7/2/79. (5683 hrs) Total shutdown period: (19129 hrs)

23) Malfunction of turbine initial pressure control. Shutdown by order of Supreme Court of 8/29 because of formal (technical) error in the licensing procedure (1536 hrs)

24) Steam leaks in the pressure suppression system. Load reduction because of repeat tests (145 hrs)

25) Faulty actuation on 2 scram accumulators (9 hrs)

UNTERWESER, 1300 MW, TC-6F-54, 1500 RPM

Commercial Operation : September 1, 1979

Operational Statistics From September 1, 1979

To December 31, 1980

	1979	1980
Power Generation, MWHRS	3,529.8	9,813.9
Availability Factor, %	93.3	89.9
Cum. Availability Factor, %	93.3	90.1
Capacity Factor, %	92.8	86.7
Cum. Capacity Factor, %	92.8	88.0

LIST OF OUTAGES BY YEAR

Cause and Duration

1979

1) Inspection of reheater (197 hrs)

1980

2) 1st refueling and maintenance outage (925 hrs)
3) Repair of HP drain piping (6 hrs)
4) False signal : Reactor trip (3 hrs)
5) Transformer disconnected for oil conditioning (5 hrs)
6) Transformer disconnected for oil conditioning (5 hrs)
7) Transformer disconnected for oil conditioning (9 hrs)

ISAR, 907 MW, TC-4F-54, 1500 RPM

Commercial Operation : March 21, 1979

Operational Statistics From March 21, 1979

To December 31, 1980

	1979	1980
Power Generation, MWHRS	4,693.0	4,396.2
Availability Factor, %	77.8	65.9
Cum. Availability Factor, %	82.6	77.5
Capacity Factor, %	68.4	57.7
Cum. Capacity Factor, %	77.7	62.3

LIST OF OUTAGES BY YEAR

Cause and Duration

1979

1) Elimination of leaks in turbine condenser (64 hrs)
2) Elimination of leaks in turbine condenser (76 hrs)
3) Incorrect operation of temporary start-up device in the area of the condensate pumps (12 hrs)
4) Leaky valves in feedwater circuit (164 hrs)
5) Leaky radial-stage throttle drain valve (3 hrs)
6) Replacement of machine transformer (699 hrs)
7) I & C change for reactor pressure vessel level measurement; circuit was not properly decoupled (10 hrs)
8) Leak elimination at cover seal of various valves (79 hrs)
9) Replacement of primary cleaning filter for generator cooling (8 hrs)
10) Logic error in the interlock of the refueling platform caused shutdown (19 hrs)

1980

11) Seal leakage on valves in feedwater, turbine area (61 hrs)
12) Defective handhole cover seal on drain tank of main steam piping (144 hrs)
13) 1st refueling and annual maintenance inspection (2341 hrs)
14) Leakage on cover seal of a relief valve (72 hrs)

15) Leaky handhole cover on reheater (61 hrs)

16) Alarm signal resulting from spray water during filling of sprinkler unit (2 hrs)

17) Leakage in weld joint in the HP area of the residual heat removal loop (301 hrs)

18) Leaky valve in control line of an isolation valve (5 hrs)

PHILIPPSBURG 1, 864 MW, TC-4F-54, 1500 RPM

Commercial Operation : February 18, 1980

Operation Data From January 1, 1980

To December 31, 1980

	1980
Power Generation, MWHRS	1,850.6
Availability Factor, %	27.3
Cum. Availability Factor, %	21.7
Capacity Factor, %	24.6
Cum. Capacity Factor, %	20.0

LIST OF OUTAGES BY YEAR

Cause and Duration

1980

1) Maintenance inspection work (324 hrs)
2) Sealing of leaks at condenser (29 hrs)
3) Sealing of leaks in seal steam leakoff system in containment structure (98 hrs)
4) Exchange of generator transformer (457 hrs)
5) Checkout of automatic turbine tester carried out as part of internal test program (14 hrs)
6) Drip leak in area of start-up line (58 hrs)
7) Faulty signal in turbine protection (9 hrs)
8) Retrofit (modification of scram system, erection of new decontamination building, replacement of feedwater line in the pressure suppression system, installation of H_2-recombination system, exchange of HP preheater & reheater-condensate cooler, etc.) per requirement of regulatory authorities (4710 hrs)