INSTRUMENTATION

3/4.3.4 TURBINE OVERSPEED PROTECTION

LIMITING CONDITION FOR OPERATION

3.3.4 At least one Turbine Overspeed Protection System shall be OPERABLE.

APPLICABILITY: MODES 1, 2*, and 3*.

ACTION:

a. With one stop valve or one control valve per high pressure turbine steam line inoperable and/or with one stop valve or one control valve per low pressure turbine steam line inoperable, restore the inoperable valve(s) to OPERABLE status within 72 hours, or close at least one valve in the affected steam line(s) or isolate the turbine from the steam supply within the next 6 hours.

b. With the above required Turbine Overspeed Protection System otherwise inoperable, within 6 hours isolate the turbine from the steam supply.

SURVEILLANCE REQUIREMENTS

4.3.4.1 The provisions of Specification 4.0.4 are not applicable.

4.3.4.2 The above required overspeed protection system shall be demonstrated OPERABLE:

a. At least once per 14 days by cycling each of the following valves through at least one complete cycle from the running position using the manual test or Automatic Turbine Tester (ATT):

1) Four high pressure turbine stop valves,
2) Four high pressure turbine control valves,
3) Four low pressure turbine stop valves, and
4) Four low pressure turbine control valves.

b. At least once per 14 days by testing of the two mechanical overspeed devices using the Automatic Turbine Tester or manual test.

c. At least once per 31 days by direct observation of the movement of each of the above valves through one complete cycle from the running position.

d. At least once per 40 months by disassembling at least one ~~of each of the above valves~~ and performing a visual and surface inspection of valve seats ~~(if applicable)~~, disks and stems and verifying no unacceptable flaws. If unacceptable flaws are found, all other valves of that type shall be inspected.

INSERT A

INSERT B

*Not applicable in MODES 2 and 3 with all main steam line isolation valves and associated bypass valves in the closed position.

9211160387 921110
PDR ADOCK 05000445
P PDR

INSERT A

high pressure turbine stop valve and one high pressure control valve

INSERT B

e. At least once per 40 months by visually inspecting the disks and accessible portions of the shafts of at least one low pressure turbine stop valve and one low pressure turbine control valve and verifying no unacceptable flaws are found. If unacceptable flaws are found, all other valves of that type shall be inspected.

ENCLOSURE 1 TO TXX-92486

Standard Review Plan, NUREG-75/087, Section 10.2

U.S. NUCLEAR REGULATORY COMMISSION

STANDARD REVIEW PLAN

OFFICE OF NUCLEAR REACTOR REGULATION

SECTION 10.2 TURBINE GENERATOR

REVIEW RESPONSIBILITIES

Primary - Power Systems Branch (PSB)

Secondary - Mechanical Engineering Branch (MEB)
Materials Engineering Branch (MTEB)
Radiological Assessment Branch (RAB)
Auxiliary Systems Branch (ASB)

I. AREAS OF REVIEW

Nuclear reactor plants include a turbine generator system (TGS) to convert the energy in steam from the nuclear steam supply system into electrical energy. The TGS consists essentially of the turbine unit and the automatic devices, alarms, and trips which control and regulate turbine action, and the generator unit and its controls. The turbine control system and the steam inlet stop and control valves, the low pressure turbine steam intercept and inlet control valves, and the extraction steam control valves control the speed of the turbine under normal and abnormal conditions, and are thus related to the overall safe operation of the plant.

The turbine generator system installed in a nuclear plant is typically equipped with redundant overspeed protection instrumentation and controls and the main steam and reheat steam control and stop valving arrangements typically provide redundancy in the valves essential for overspeed protection. The intent of the review under this SRP section is to verify that such redundancy, in conjunction with inservice inspection and testing of the essential valves, makes a turbine overspeed condition above the design overspeed very unlikely. Assessment of the risk to essential plant systems and structures from potential turbine missiles is reviewed under SRP Section 3.5.1.3.

1. The PSB reviews the turbine generator system and the components and subsystems normally provided with this equipment with respect to the following considerations:

   a. The general arrangement of the turbine and associated equipment with respect to safety-related structures and systems and balance of plant.

   b. The types and locations of main steam stop and control valves, reheat stop and intercept valves, and associated piping arrangements.

USNRC STANDARD REVIEW PLAN

Standard review plans are prepared for the guidance of the Office of Nuclear Reactor Regulation staff responsible for the review of applications to construct and operate nuclear power plants. These documents are made available to the public as part of the Commission's policy to inform the nuclear industry and the general public of regulatory procedures and policies. Standard review plans are not substitutes for regulatory guides or the Commission's regulations and compliance with them is not required. The standard review plan sections are keyed to Revision 2 of the Standard Format and Content of Safety Analysis Reports for Nuclear Power Plants. Not all sections of the Standard Format have a corresponding review plan.

Published standard review plans will be revised periodically, as appropriate, to accommodate comments and to reflect new information and experience.

Comments and suggestions for improvement will be considered and should be sent to the U.S. Nuclear Regulatory Commission, Office of Nuclear Reactor Regulation, Washington, D.C. 20555.

c. The capability of the turbine generator control and overspeed protection systems to detect a turbine overspeed condition and to actuate appropriate system valves or other protective devices to preclude an overspeed condition above the design overspeed.

d. The overspeed protection instrumentation and controls with respect to redundancy, testability and reliability.

2. The PSB reviews the inservice inspection and operability assurance program for valves essential for overspeed protection.

3. The PSB reviews the applicant's proposed technical specifications for operating license applications as they relate to areas covered in this SRP section.

Secondary reviews are performed by other branches and the results used by the PSB to complete the overall evaluation of the system. The secondary reviews are as follows: the ASB determines that appropriate seismic and quality group classifications have been established for system components where appropriate. The ASB and MEB determine that the TGS is in accordance with Branch Technical Positions ASB 3-1 and MEB 3-1 as related to pipe cracks or breaks in high and moderate energy piping systems outside of containment. The MEB confirms that the components, piping, and structures are designed in accordance with applicable codes and standards. The MTEB verifies that inservice inspection requirements are met for system components, and will verify the compatibility of the materials of construction with service conditions. The RAB determines if any radiation shielding is necessary to assure safe access to turbine equipment.

II. ACCEPTANCE CRITERIA

There are no general design criteria or regulatory guides that are directly applicable to the design evaluation of the turbine generator. Acceptability of the design of the turbine generator system, as described in the applicant's Safety Analysis Report (SAR), is based on the specific criteria listed below and on the similarity of the design to that of plants previously reviewed and found acceptable.

1. A turbine control and overspeed protection system should be provided to control turbine action under all normal or abnormal operating conditions, and to assure that a full load turbine trip will not cause the turbine to overspeed beyond acceptable limits. Under these conditions, the control and protection system should permit an orderly reactor shutdown either by use of the turbine bypass system and main steam relief system or other engineered safety systems. The overspeed protection system should meet the single failure criterion and should be testable when the turbine is in operation.

2. Turbine main steam stop and control valves and reheat steam stop and intercept valves should be provided to protect the turbine from exceeding set speeds and to protect the reactor system from abnormal surges. The reheat stop and intercept

valves should be capable of closure concurrent with the main steam stop valves, or of sequential closure within an appropriate time limit, to assure that turbine overspeed is controlled within acceptable limits. The valve arrangements and valve closure times should be such that a failure of any single valve to close will not result in excessive turbine overspeed in the event of a TGS trip signal.

3. The extraction steam check valves provided at extraction connections shall be capable of closing within an appropriate time limit to maintain stable turbine speeds in the event of a TGS trip signal.

4. The TGS should be provided with the capability to permit periodic testing of components important to safety while the unit is operating at rated load.

5. The inservice inspection program for main steam and reheat valves should include the following provisions:

   a. At approximately 3-1/3-year intervals, during refueling or maintenance shutdowns coinciding with the inservice inspection schedule required by Section XI of the ASME Code for reactor components, at least one main steam stop valve, one main steam control valve, one reheat stop valve, and one reheat intercept valve should be dismantled and visual and surface examinations conducted of valve seats, disks, and stems. If unacceptable flaws or excessive corrosion are found in a valve, all other valves of that type should be dismantled and inspected. Valve bushings should be inspected and cleaned, and bore diameters should be checked for proper clearance.

   b. Main steam stop and control valves and reheat stop and intercept valves should be exercised at least once a week by closing each valve and observing by the valve position indicator that it moves smoothly to a fully closed position. At least once a month, this examination should be made by direct observation of the valve motion.

6. Unlimited access to all levels of the turbine area under all operating conditions should be provided. Radiation shielding should be provided as necessary to permit access.

7. Connection joints between the low pressure turbine exhaust and the main condenser should be arranged to prevent adverse effects on any safety-related equipment in the turbine room in the event of rupture (it is preferable not to locate safety-related equipment in the turbine room).

8. Branch Technical Positions ASB 3-1 and MEB 3-1 should be used to determine the acceptability of the effects of postulated TGS piping failures on safety-related equipment.

9. Regulatory Guide 1.68 should be used with regard to preoperational and startup testing of the power conversion system.

For those areas of review identified in subsection I of this SRP section as being the responsibility of other branches, the acceptance criteria and their methods of application are contained in the SRP sections corresponding to those branches.

III. REVIEW PROCEDURES

The procedures below are used during the construction permit (CP) review to determine that the design criteria and bases and preliminary design as set forth in the Preliminary Safety Analysis Report meet the acceptance criteria given in subsection II. For review of operating license (OL) applications, the procedures are utilized to verify that the initial design criteria and bases have been appropriately implemented in the final design as set forth in the Final Safety Analysis Report.

The review procedures for OL applications include a determination that the content and intent of the technical specifications prepared by the applicant are in agreement with the requirements for system testing, minimum performance, and surveillance developed as a result of the staff's review.

The review procedures given are for a typical turbine generator system. Any variance of the review, to take account of a proposed unique design, will be such as to assure that the system meets the criteria of subsection II. The reviewer evaluates the TGS, subsystems, and components of the unit that are considered essential for the safe integrated operation of the reactor facility. The reviewer will select and emphasize material from this SRP section as may be appropriate for a particular case.

Upon request from the primary reviewer, the secondary review branches will provide input for the areas of review stated in subsection I. The primary reviewer obtains and uses such input as required to assure that this review procedure is complete.

1. The SAR is reviewed to determine that the system description and piping and instrumentation diagrams (P&IDs) show the turbine generator system. The general arrangement of the TGS and associated equipment with respect to safety-related structures, systems, and components is noted.

2. The reviewer verifies the adequacy of the control and overspeed protection system and determines that:

   a. Support systems, subsystems, control systems, and alarms and trips will function for all abnormal conditions, including a single failure of any component or subsystem, and will preclude an unsafe turbine overspeed. The indepth defense that is provided by the turbine generator protection system to preclude excessive overspeeds should be designed with diverse protection means.

b. For normal speed-load control, the speed governor action of the electro-hydraulic control system fully cuts off steam at approximately 103 percent of rated turbine speed by closing the control, stop, and intercept valves.

c. A mechanical overspeed trip device is provided that will actuate the control, stop, and intercept valves at approximately 111 percent of rated speed.

d. An independent and redundant backup electrical overspeed trip circuit is provided that senses the turbine speed by magnetic pickup and closes all valves associated with speed control at approximately 112 percent of rated speed. This backup electrical overspeed trip system may utilize the same sensing techniques as the electro-hydraulic control system. However, the circuitry is reviewed to determine that the control signals from the two systems are isolated from and independent of one another.

3. The main steam stop and control and the reheat stop and intercept valving arrangements and valve closure times are reviewed to ensure that no single valve failure can disable the overspeed control function.

4. The extraction steam valving arrangements and valve closure times are reviewed to see that stable turbine operation will result after a TGS trip.

5. The capability for testing of essential components during TGS operation is reviewed.

6. The proposed i service inspection program for essential speed control valves is reviewed to verify that it includes the provisions of item 5 of subsection II.

7. The reviewer consults with RAB with regard to expected radiation levels around the TGS and the degree of access to TGS components during operation.

8. If there are safety-related systems or portions of systems located close to the TGS, the physical layout of the system is reviewed to assure that protection has been provided from the effects of high and moderate energy TGS piping failures or failure of the connections from the low pressure turbine section of the main condenser. The means of providing such protection will be given in Section 3.6 of the SAR, and the procedures for reviewing this information are given in the corresponding SRP sections.

IV. EVALUATION FINDINGS

The reviewer verifies that sufficient information has been provided and his review supports conclusions of the following type, to be included in the staff's Safety Evaluation Report:

"The turbine generator system includes all components and equipment normally provided including turbine main steam stop and control valves and reheat steam stop and intercept valves. The scope of review of the turbine generator system for the _______ plant included layout drawings, piping and instrumentation diagrams, and descriptive information for the system and for control and supporting systems that are essential to its operation. [The review has determined the adequacy of the applicant's proposed design criteria and bases for the turbine generator system and the requirements for safe operation of the system during normal, abnormal, and accident conditions (CP).] [The review has determined that the design of the turbine generator system and supporting systems is in conformance with the design criteria and design bases (OL).]

"The basis for acceptance in the staff review has been conformance of the applicant's designs, design criteria, and design bases for the turbine generator system and supporting systems to applicable staff technical positions and industry standards.

"The staff concludes that the design of the turbine generator system conforms to all applicable staff positions and industry standards, and is acceptable."

V. REFERENCES

1. Branch Technical Positions ASB 3-1, "Protection Against Postulated Piping Failures in Fluid Systems Outside Containment," attached to SRP Section 3.6.1, and MEB 3-1, "Postulated Break and Leakage Locations in Fluid System Piping Outside Containment," attached to SRP Section 3.6.2.

2. Regulatory Guide 1.68, "Initial Test Programs for Water-Cooled Reactor Power Plants."

ENCLOSURE 2 TO TXX-92486

Siemens letter to Mr. R. T. Jenkins from Mr. P. C. Hosbein,
dated February 14, 1992

SIEMENS

Post-It™ brand fax transmittal memo 7671	# of pages ▸ 5
To R.T. Jenkins	From Pete Hosbein
Co. TU Electric	Co. SPC
Dept.	Phone #
Fax #	Fax #

February 14, 1992

Mr. R.T. Jenkins
Manager, System Engineering
TU Electric
Comanche Peak SES
P.O. Box 2300
Glen Rose, TX 76043

Re: Design and Inspection of Main Turbine Valves
Reference letter from P.C. Hosbein to
J.J. Kelley, dated January 23, 1990

Dear Mr. Jenkins:

This letter is in response to TU Electric's request for information regarding the design and inspection of the High Pressure (HP) and Low Pressure (LP) stop and control valves at the Comanche Peak Steam Electric Station Units 1 and 2.

<u>HP Valve Design and Construction</u>

Th HP stop and control valves are designed to control the speed and load of the main turbine-generator. The HP stop valves are designed for extremely fast closing to isolate the turbine preventing it from reaching a potentially destructive overspeed condition.

The HP stop and control valves are combined in a common body and the primary components are the seat, cone and stem, with an actuator that controls the position of the cone. The seat of the valve is machined from bar stock, material 21CrMoN.N47. The seat has a stellite overlay in the area of contact to the cone. After machining the seat is PT inspected. The cone is a casting GS-C256 also with various overlays in different areas. The cone is also PT inspected as part of the manufacturing process.

The stem of the valve is made of 34CrAlMo5 material with a gas nitrade surface hardening. The valve stem is also PT inspected during manufacture.

In general when ever there is a special surface hardening or coating process, (i.e. flame spray or gas nitride) the surface of the material where the process is applied is PT inspected.

Siemens Power Corporation

P.O. Box 180 Bradenton, Florida 34206 (813) 723-4100 FAX (813) 723-4210

LP Valve Design and Construction

The LP stop and control valves are butterfly valves, the primary components are the flapper, the shaft, and the pipe, which contains the hot reheat steam.

The flapper is a casting which is fabricated in one piece from 18CrMo910 steel. The flapper rotates within the pipe 90° in an open or closed position. The flapper does not make metal to metal contact in the closed position. There is a 2mm gap between the edge of the flapper and the inside pipe wall. There is no seat as such in this butterfly valve. A surface crack examination is made of the flapper in the shop, since this is a casting an MT is performed. A hardness test is also performed in the shop.

The shaft is made from a solid piece of bar stock material type 21CrMcNiV47 (American equivalent ASTM A 193 Gr816). The shaft is inserted through the flapper and secured with 6 locking pins. The assembly is performed at the factory and the shaft has never been removed for any reason at any operating plant. Although it is physically possible the work would be quite extensive and dangerous. The factory performs a surface crack examination of the shaft using a PT method. A run out check is also made after machining.

The 48" pipe that surrounds the flapper is carbon steel material. The welds that were made to the hot reheat elbows were RT inspected during construction.

Normal Inspection Recommendations

Please refer to my letter dated January 23rd 1990 to Mr. James J. Kelley (copy attached). The HP stop and control valves normally have been PT inspected to check for cracks or surface indictions in the stem cone and seat.

During an outage on the steam turbine Siemens recommends that one LP stop valve and one LP control valve be inspected. This inspection calls for only visual examination of the flapper from inside the hot reheat pipe and a visual examination of the shaft after the steam seals and bearings have been removed. If any findings are made during this examination further testing would be recommended.

HP Valve Historical Information

The simplicity of individual, symetrical casing valves has the advantages of separate valve seats and separate control functions. This design has been responsible for an

operational history of high reliability and excellent operating performance since its adoption in 1958.

LP Valve Historical Information

Siemens has steam Turbines at 32 nuclear plants around the world. All these steam turbines have butterfly valves of similar construction to the valves at Comanche Peak The normal inspections as described above have been performed through the 18 years that the nuclear steam turbines have operated.

No further inspection beyond the normal visual inspection have been performed. These butterfly valves have performed without problems on the shaft or flappers at all the the nuclear steam turbines around the world.

We do not anticipate changing the examination in the future because of the good performance of this design. There is no metal to metal contact on these butterfly valves that would cause stresses or strains to the material. The temperature and pressure of the hot reheat steam is relatively low and no metal creep on metal fatigue of the flapper or shaft can occur.

We will provide additional factory testing and backup documentation if requested.

Sincerely,

Peter C. Hosbein
Manager, Service Development
Product Service Division

PCH/umy

Attachment

cc: W.J. Cahill
A.B. Scott
J.J. Kelley

ENCLOSURE 3 TO TXX-92486

Utility Power Corporation letter to Mr. J. J. Kelley
from P. C. Hosbein, dated January 23, 1990

SD

Utility Power Corporation

4800 Spring Valley Road, Suite 205, Dallas, Texas 75244 / (214) 960-8885

January 23, 1990

Mr. James J. Kelley, Jr.
Plant Operations Manager 009
Comanche Peak SES
TU Electric Company
P.O. Box 2300
Glen Rose, TX 76043

Re: Non-destructive Examination of High Pressure and Low Pressure Turbine Valves.

Dear Jim:

We inspected one of each of the HP and LP stop and control valves per the surveillance requirements, section 4.3.4.2 of the FSAR this past Summer. We would like to clarify our current recommendations for non-destructive examination (NDE) of the valves for your future maintenance plans.

High Pressure Turbine Stop Valves

Valve Cone (Disk): Visually inspect for wear, chipping, deformation, corrosion, erosion, cracking and scaling.
Valve Seat: Visually inspect for wear, chipping and cracking. Perform contact check with cone. Liquid penetrant (PT) for cracks.
Valve Body, Bonnet and Flange: Visually inspect for scaling, cracking, erosion and corrosion. If necessary, perform magnetic particle test (MT).
Valve Spindle (Stem): Visually inspect for bending, distortion, galling and cracking. Perform runout check using "V" blocks. Perform 100% PT for cracks of hardened surface.
Valve Bushing: 100% dimensional checks.

High Pressure Turbine Control Valves

Perform VT, MT and PT as per HP stop valve.

Low Pressure Turbine Stop Valves

Since there is no "seat" on these butterfly-type valves, a seat inspection is superfluous.

Valve Disc (Butterfly): Visual inspection (VT) for corrosion, erosion and cracking.

Utility Power Corporation

Seals and Seal Bushings: VT for corrosion, erosion and cracking.

Shaft and Shaft Bearings: VT for corrosion, erosion and cracking. 100% PT of the babbitted surface and 100% UT of the babbitt to base metal bond.

Low Pressure Turbine Control Valves

Perform VT as per LP stop valves.

These NDE recommendations will be valid any time an HP or LP valve is inspected during an outage. The work scope for turbine generator work during refueling outages should be determined at least 18 months prior to the outage to allow sufficient time for parts delivery and planning.

Together, Utility Power Corporation and TU Electric Company should begin planning for the first refueling outage on Unit #1, scheduled for 1991. I would be happy to meet with you or your representatives at the earliest convenient time.

Very truly yours,

Peter Hosbein

Peter C. Hosbein
Manager, Service Development

cc: M.W. Cunningham
D.L. Davis
G.E. Jergins
F.W. Madden
A.B. Scott, Jr.
I.C. Whitt
B.W. Wieland

bcc: P.K. Bhavnani
A.E. Clayton
R.F. Janoske
J.J. Rauter

PCH:lm

ENCLOSURE 4 TO TXX-92486

Allis-Chalmers Powers Systems, Inc.

Engineering Report No. ER-504

"PROBABILITY OF TURBINE MISSILES
from 1800 r/min Nuclear Steam Turbine-Generators
with 44-inch Last Stage Blades," October 1975

9211160 ~~372~~ 100pp.

CONTENTS

CONTENTS

1. INTRODUCTION

This report provides information on the probability of occurrence of turbine missiles from burst-type failure of low pressure (LP) blade disks of 1800 r/min turbines designed for nuclear power plant applications. Two distinct cases are covered: 1) LP disk failure at or near design speed (≤120% of rated speed), and 2) LP disk burst due to an excessive overspeed incident.

For the first case, information is provided on the turbine disk integrity including details of design, manufacture and quality assurance which provide very high reliability against disk failure in the design speed range. In addition, it should be noted that even if a disk burst occurs in the design speed range, the disk fragments would be contained by the turbine casings. The details of this analysis are given in A-CPSI Engineering Report #ER-503 (1) which shows that burst speeds in excess of about 160% of rated would be required to yield external turbine disk missiles. Since an LP turbine disk missile can only penetrate the turbine casings at a turbine-generator speed in excess of about 160%, the probability of significant turbine disk missiles within the design speed range of 120% is considered zero.

For the second case, LP turbine disk missiles with the highest energy level are defined for a 180 to 185% speed range, where the average tangential disk stress is equal to 85% of the disk material ultimate strength. The probability of turbine missiles for this case is assumed to be equal to the probability of an overspeed incident of greater than 120% of rated speed. This is determined by means of a reliability analysis of the turbine valves, speed control and overspeed trip systems. The results of reliability analysis

are values of 1.6×0^{-7} per unit-year for a 6-flow turbine-generator and 2.1×10^{-7} per unit-year for a 4-flow turbine-generator. These values are intended for use as "P_1" in the equation $P = P_1 \times P_2 \times P_3$ where P = overall probability of turbine missile damage, P_2 = striking probability, and P_3 = damage probability. Information needed to calculate P_2 and P_3 is given in A-CPSI Engineering Report #ER-503.

2. HISTORICAL DATA

Historical data on actual cases of turbine-generator rotor failures is of general interest in connection with the question of probability of turbine missiles. However, as discussed below, such data is not directly applicable, for modern turbines and should not be used for predicting future probabilities of missiles.

A study published as Reference No. (2) includes data on turbine and generator rotor failures covering approximately 70,000 unit-years of operating experience of units larger than 50 Mw from 1950 to 1972. This study reported a total of 14 failures which are as listed in Table 1. Essentially all of these failures are not applicable to the turbine missile probability considered herein for the following reasons:

1. Most of the failures were primarily due to high nil-ductility transition temperatures and low fracture toughness, presence of hydrogen flakes or non-metallic inclusions and relatively undeveloped quality assurance procedures, all of which were characteristic of steel melting practice and forging technology prior to the mid-1950's. Since then, major improvements including introduction of vacuum degassing for all alloy steel grades used for turbine and generator forgings and application of sophisticated ultrasonic testing techniques have greatly reduced the possibility of failure due to these causes.

Table 1 Failures of Steam Turbine-Generator Rotors at or Near Operating Speeds

Units Larger than 50 Mwe from 1950 to 1972

Size, MWe		Manufacturer & other Identification	Year Failed	Type of Failure	Suspected Cause of Failure	Comment
1.	63	Siemens	1951	Low pressure turbine rotor burst	Brittle fracture	Missiles-factory test
2.	100	GE (Tanners Creek #1)	1953	Low pressure turbine 1st stage disk broke	High temp. rupture	No external missiles
3.	100	GE (Arizona Pub. Service)	1954	Gener :or rotor burst	Brittle fracture	Missiles-factory test
4.	168	GE (Cromby #1, Phila.Elec.)	1954	Generator rotor burst	Brittle fracture through repair	No external missiles
5.	100	Charles A. Parsons	1954	Generator retaining ring burst	Brittle fracture thru vent holes	External missiles
6.	100	Charles A. Parsons	1954	Generator retaining ring burst	Brittle fracture t ru vent holes	Limited external missiles
7.	150	A-C (Commonwealth Edison)	1954	Turbine spindle burst	Brittle fracture	External missiles
8.	153	GE (PG&E Pittsburg #1)	1956	Generator rotor burst	Brittle failure	No missiles
9.	87	Hinkley Pt. A #5	1969	Disks failed	Brittle failure	Missiles
10.	87	Hinkley Pt. A #4 & #6	1970 1970	Disks failed in pit	Brittle failure	Missiles-factory test
11.	?	Mitsubishi (ENESA Spain) Westinghouse design	1970	Low pressure turbine rotor burst	Flawed forging (?)	Missiles-factory test
12.	150	GE (Cutler #6 FP&L)	1969	Generator field winding assembly failure	Out-of-step with system	No energetic missiles
13.	105	GE (Essex #1 PSE&G)	1972	Generator field failure	Abrupt b.aking of generator	Medium energy missiles-shaft coupling
14.	?	Mitsubishi (Kainan) Westinghouse design	1972	Rotor failure during overspeed test in plant	(?)	External missiles

2. Seven of 14 failures in Table 1 were of generator rotors which are not directly applicable to LP disk failure probability.

3. Four of 14 failures in Table 1 occurred in factory test pits which are not directly applicable to failures in power plants.

4. Several failures in Table 1 produced no significant missiles, and are therefore not directly applicable to failure probability for purposes of turbine missile analysis.

Historical data from A-CPSI's parent companies is included in Table 1 and the overall study in Reference (2). This specific experience, updated through the end of 1974 and expanded to include units larger than 10 Mw is given in Tables 2 and 3.

Based on the data in Table 2 for KWU, Siemens and AEG units larger than 10 Mw from 1950 through 1974, there was only one rotor failure during a factory test in a total of 8,688 unit-years (26,187 rotor-years) of operating experience. The failure was a brittle fracture of an LP turbine rotor in 1951. The cause of this failure was primarily due to excessively high hydrogen content of the forging.

Table 2 Rotor Failures in Kraftwerk Union, Siemens and AEG Steam Turbine-Generators larger than 10 Mw put in Operation since 1950

Year	No. of Unit Put in Operation	No. of Rotors Put in Operation	Cumulative Unit-Years of Service	Cumulative Rotor-Years of Service	Number of Rotor Failures
1950	10	28	5	14	0
1951	9	26	20	55	1 (1)
1952	21	52	[illegible]	135	0
1953	19	53	100	268	0
1954	32	90	175	472	0
1955	20	57	276	750	0
1956	47	121	411	1117	0
1957	41	124	590	1606	0
1958	49	150	814	2232	0
1959	24	91	1074	2979	0
1960	29	92	1361	3817	0
1961	26	90	1675	4745	0
1962	30	91	2017	5766	0
1963	41	131	2395	6897	0
1964	33	112	2704	8149	0
1965	25	95	3254	9505	0
1966	39	108	3741	10,962	0
1967	38	118	4246	12,532	0
1968	33	97	4797	14,210	0
1969	36	118	5385	15,995	0
1970	34	112	6002	17,895	0
1971	21	81	6650	19,892	0
1972	13	38	7315	21,948	0
1973	15	41	7994	24,044	0
1974	15	53	8688	26,187	0
Totals	701	2169	8688	26,187	1

NOTES:

1. Brittle facture of an LP turbine rotor during factory overspeed test.

2. Includes all turbine and generator rotors, but not exciter rotors.

Table 3 Rotor Failures in Allis-Chalmers Steam Turbine-Generators Larger than 10 Mw put in Operation since 1950

Year	No. of Units Put in Operation	No. of Rotors Put in Operation	Cumulative Unit-Years of Service	Cumulative Rotor-Years of Service	Number of Rotor Failures
1950	8	22	4	11	0
1951	8	18	16	42	0
1952	5	12	34	88	0
1953	11	25	60	152	0
1954	7	23	95	240	1 (1)
1955	8	24	138	352	0
1956	9	12	189	482	0
1957	6	14	248	625	0
1958	20	61	320	805	0
1959	8	23	406	1027	0
1960	11	37	501	1279	0
1961	10	22	607	1566	0
1962	5	4	720	1876	0
1963	5	17	838	2201	0
1964	6	20	962	2545	0
1965 (2)	1	3	1089	2900	0
1966	-	-	1217	3257	0
1967	-	-	1345	3614	0
1968	-	-	1473	3971	0
1969	-	-	1601	4328	0
1970	-	-	1729	4685	0
1971	-	-	1857	5042	0
1972	-	-	1985	5399	0
1973	-	-	2113	5756	0
1974	-	-	2241	6113	0
Totals	128	357	2241	6113	1

NOTES:

1. Brittle fracture of an LP turbine rotor in a power plant.

2. Allis-Chalmers stopped taking orders for steam turbine-generators in 1962 and put last unit in operation in 1965, but re-entered the business together with Kraftwerk Union forming Allis-Chalmers Power Systems, Inc., in 1970.

As shown in Table 3 for Allis-Chalmers units larger than 10 Mw from 1950 through 1974, there was one rotor fa..ure in a total of 2241 unit-years (6113 rotor-years) of operating experience. The failure was a brittle fracture of an LP turbine rotor in a power plant during an overspeed-trip test, and is discussed in detail in Reference (3). This failure also involved high hydrogen content in the forging.

The combined experience of all parent companies of A-CPSI from 1950 to 1974 is therefore two failures and a total of 10,929 unit years (32,300 rotor-years) of experience.

As discussed i: connection with Table 1, the factory test pit failure of Siemens and the failure of an Allis-Chalmers rotor involving high H_2 content in the forging are not directly applicable to the turbine missile question.

However, it is significant that there have been no turbine rotor or disk failures in a power plant for units built by Siemens and KWU who are the primary sourcos of design and quality assurance of units provided by A-CPSI.

3. TURBINE DISK INTEGRITY

The following is information on those aspects of design, manufacture and quality assurance of 1800 rpm LP turbine disks and rotors which provide a very high degree of safety against burst-type failure of the disks.

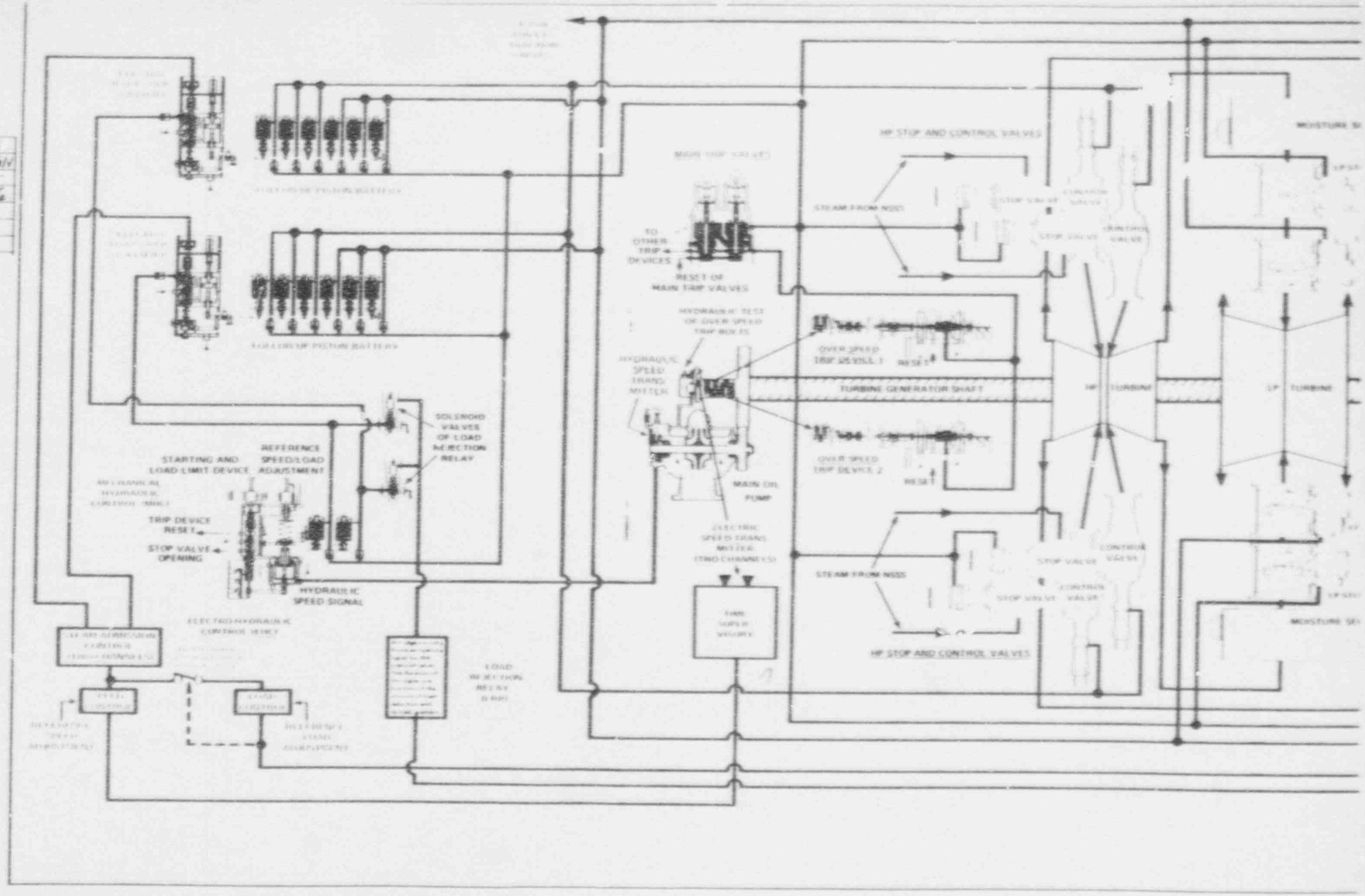
3.1 Design

The configuration of an 1800 r/min LP turbine with 44-inch last stage blades is shown in Drawing TY 5.01. Each two flow LP turbine rotor is made from a stepped shaft with a total of 10 shrunk-on blade disks arranged in symmetrical groups of 5. The material for disks 1, 2, 4 and 5 has the German standard designation of 26 NiCrMoV145 which is a 3.5% Ni alloy steel similar to ASTM A-471. Disk 3 is made from a similar alloy except with 2.9% Ni which is designated 26 NiCrMoV115.

	26 NiCrMoV145	26 NiCrMoV115
Nominal Chemical Composition in %:	3.5 Ni, 1.50 Cr	2.9 Ni, 1.50 Cr
	0.26 C, 0.30 Mn	0.26 C, 0.30 Mn
	and max 0.15 V	and max 0.15 V

Mechanical Properties (max value for tensile strength, all others min values)

At 68°F (20°C):	26 NiCrMoV145	26 NiCrMoV115
Tensile Strength: Ksi	156	149
0.2% Offset Yield Strength, Ksi	128-135	114-121
Elongation (L/d = 5), %	15	16
Reduction of Area, %	40	50
Notch Impact Strength at -4F, Ft-lb (Average from 3 Charpy-V specimens)	35	35
NDT-Temp., °F max	-76	-58



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The compressive residual stress level of the heat-treated disk forgings, as measured by the ring core process (see Reference 4) shall not exceed 11,500 psi. Residual tensile stresses are not permitted.

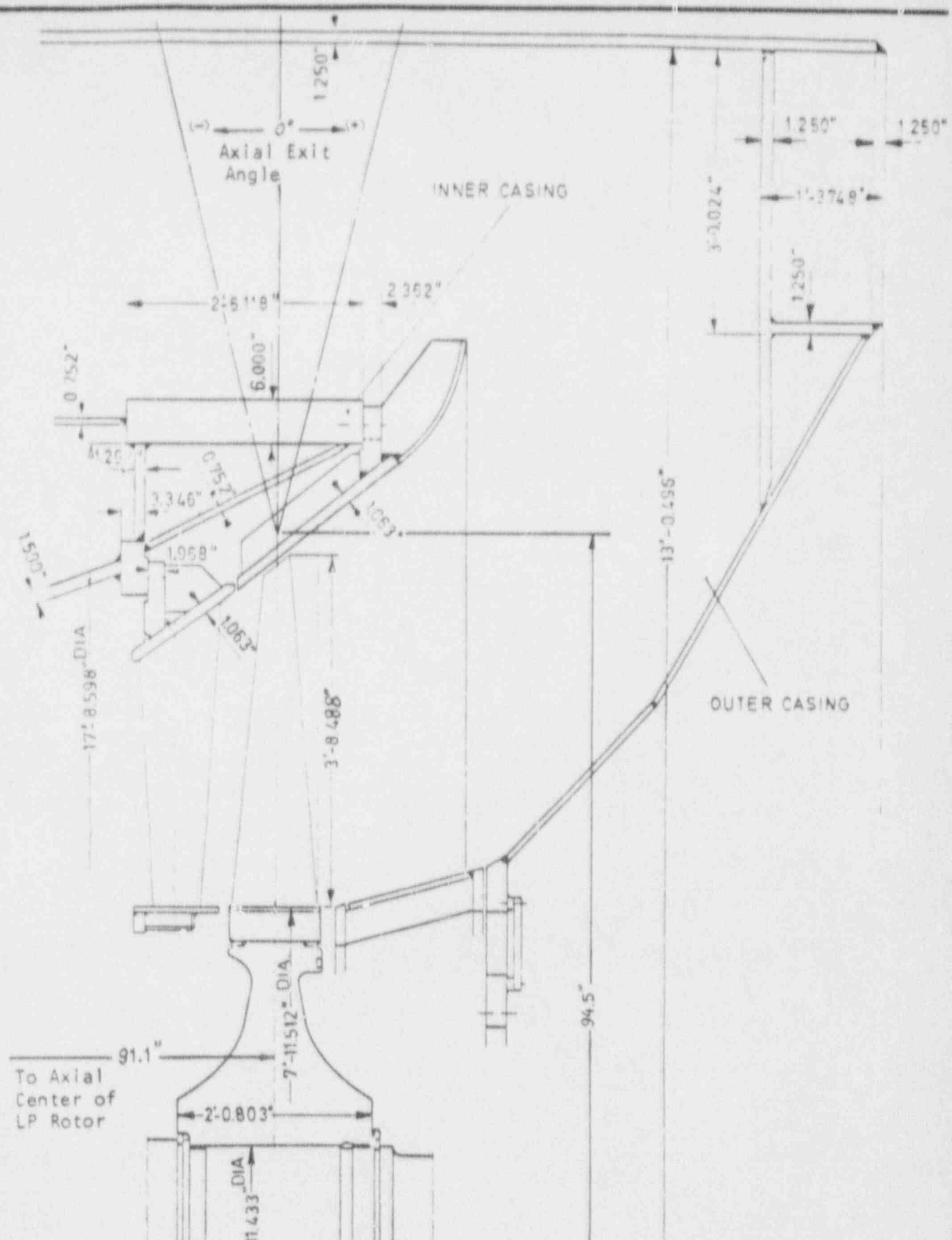
The dimensions in the last stage region, and the dimensions of the last stage disk are given in Drawings MA 4.04a and MA 4.05 respectively. The average tangential stress in the last stage disk is plotted as a function of speed in Figure MA 4.16. It can be seen that at speeds up to 120% of rated, the maximum disk stress at the shrink fit is less than one-half of the burst strength of the material. Thus, disk failure in this speed range could only occur if the material is seriously defective or if a major error is made in design or manufacturing.

3.2 Safety Analysis of LP Disks

The safety analysis of each disk design is based on the principles of Linear Elastic Fracture Mechanics (LEFM). For the purpose of analysis of the inner portion of the disk (see Figure MA 4.17), it is assumed that a flaw of 5 mm diameter equivalent flaw size (as determined by the Krautkraemer ultrasonic test method) exists at the worst possible (highest stressed) location. In addition, it is assumed that the flaw with this stated flaw area has the "worst case flaw geometry" as defined by LEFM theory. This is a long shallow elliptical crack having a depth-to-length ratio of ≤ 0.2.

The hypothetical growth of the assumed worst case-flaw is then calculated for an assumed number of about 4400* full stress cycles over the expected

* In the case of disks, approximately 4400 cycles (start-ups) are assumed; however, the general design of turbines of this type is based on approximately 2600 cycles.



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Last Stage Region
of an 1800 r/min Turbine
with 44" LSB's

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MA 4.04 a

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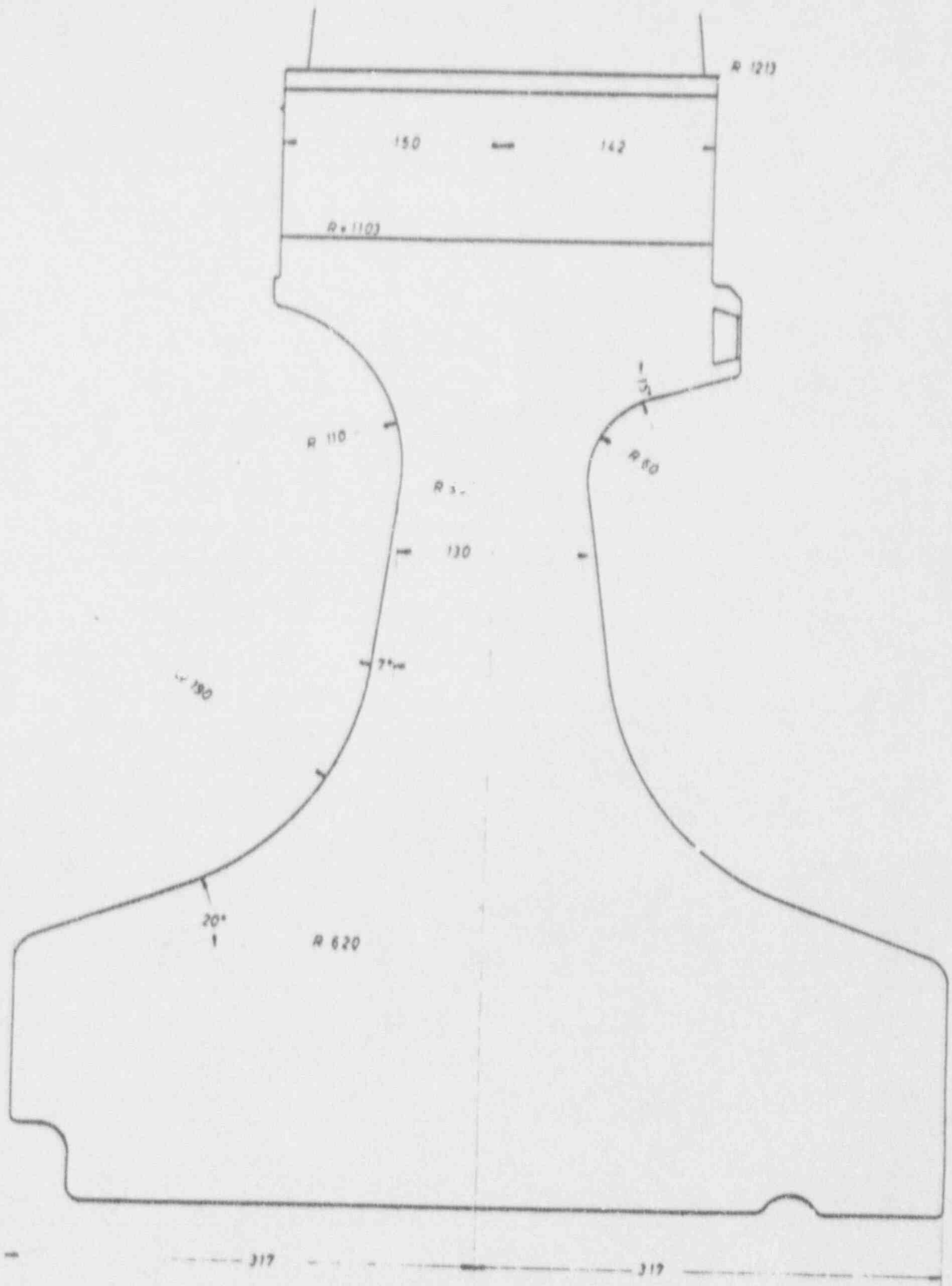

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NOTE: All dimensions are in millimeters.

Last Stage Disc
of an 1800 r/min Turbine
with 44" LSB's

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22.3.74 Roettger TOM

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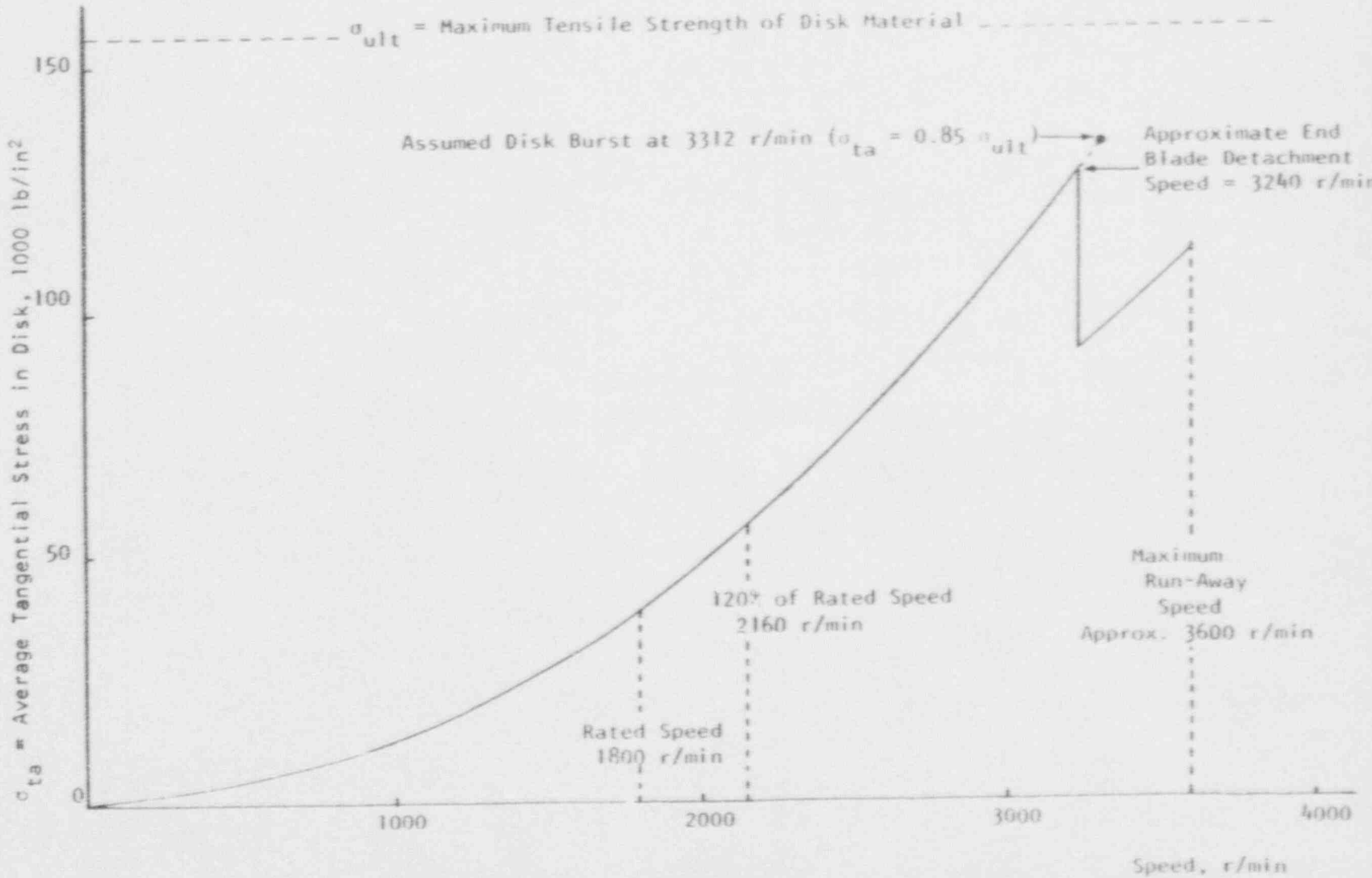

Average Tangential Stress vs Speed
Last Stage LP Turbine Disk
with 44-inch LSB's

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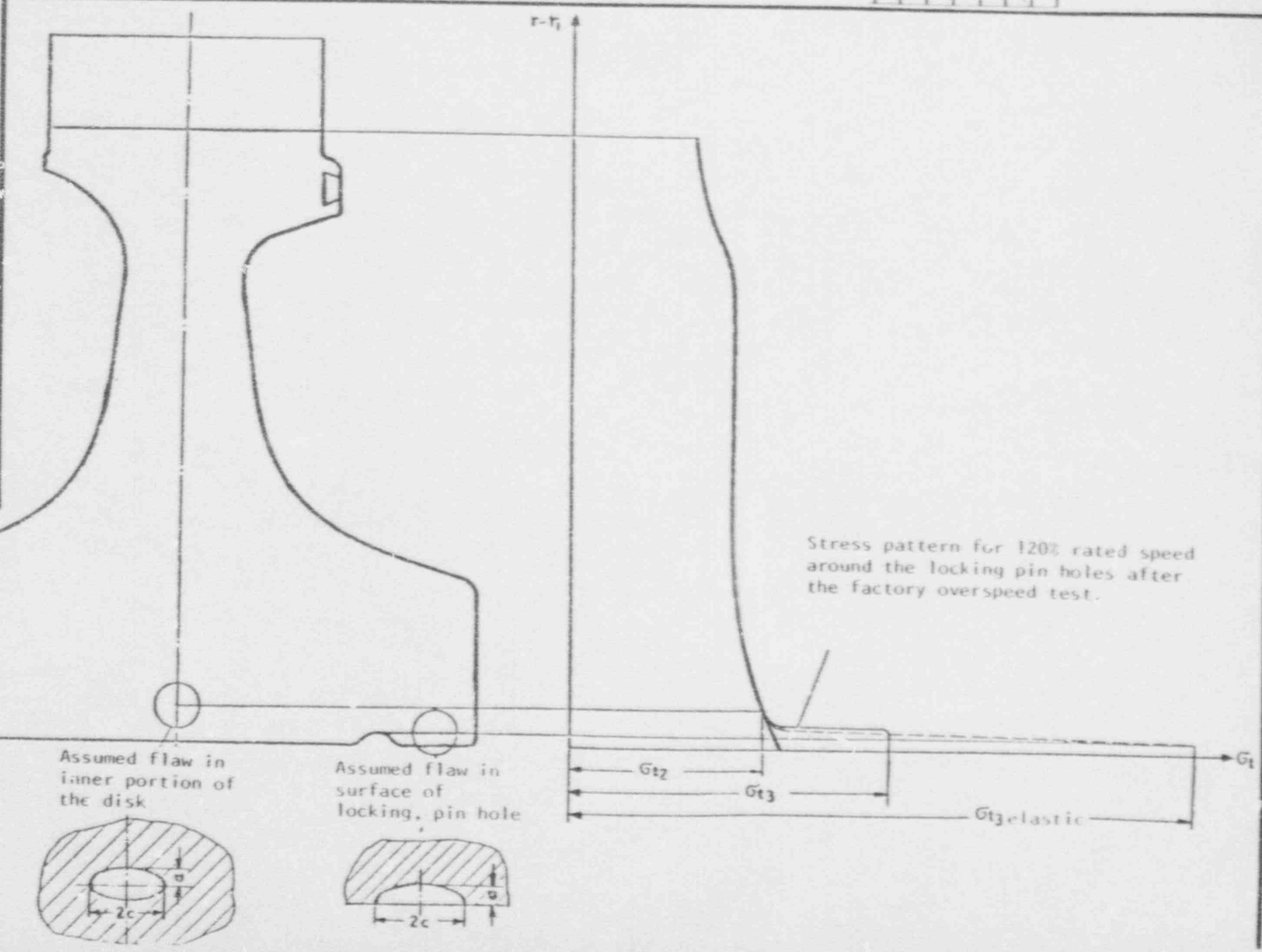


Information to Safety Analysis of LP Disks

ALLIS-CHALMERS POWER SYSTEMS, INC.

MA 4.17

life of the unit. The design requirement is that the calculated crack depth at the end of the unit's lifetime must remain below 50% of the critical crack depth, a_{cr}, for 120% of rated speed. The crack growth rate is calculated by the following general relationship:

$$\frac{da}{dN} = C\Delta K^m \left(\frac{10^{-6}\ mm}{Cycles}\right)$$

where:

a = crack depth (mm)

N = number of cycles

ΔK = cyclic stress intensity range at crack tip ($kp/mm^{3/2}$)

$\Delta\sigma$ = applied stress range (kp/mm^2)

C = a constant

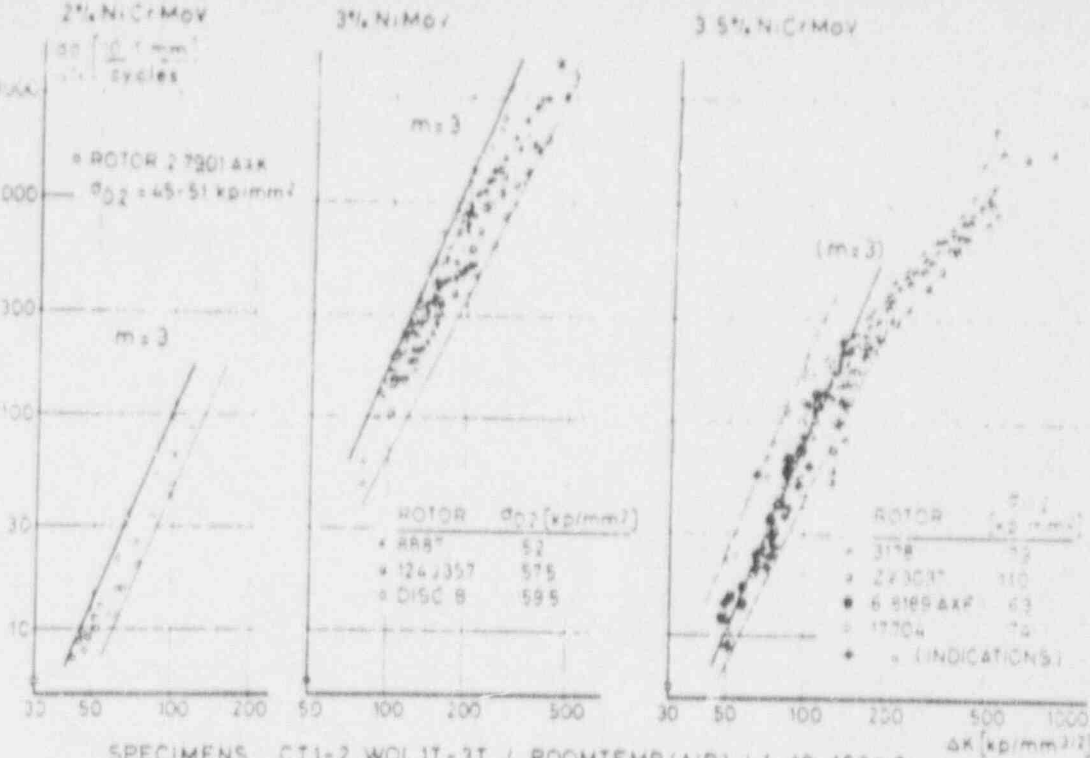
m = the fatigue crack growth exponent of the disk material

Q = crack shape parameter

$g(a)$ = crack geometry factor ($\sqrt{mm}$)

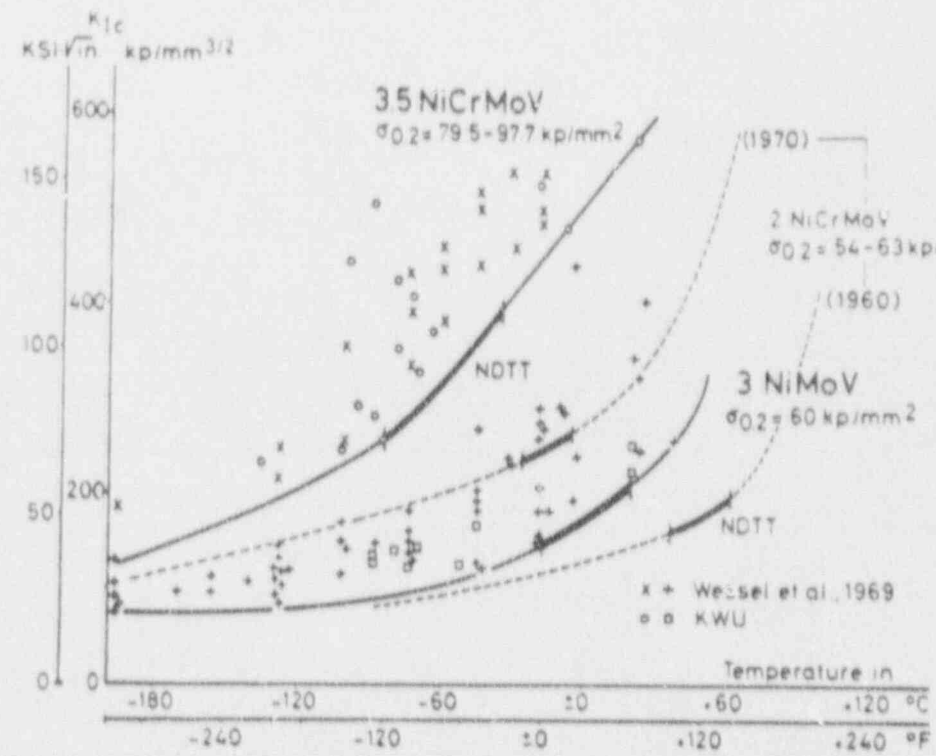
Fatigue crack growth data for different rotor and disk materials are obtained in extensive laboratory measurements as discussed in References (5) and (6). A few examples are illustrated in Figure MA 4.15a. For the safety analysis of disks, the upper boundary curves of the fatigue crack growth rate scatterbands are used in the calculations.

The cyclic stress intensity range, ΔK, at the tip of the crack is calculated by LEFM methods. The ΔK-range is linearly proportional to the applied stress range, $\Delta\sigma$, and depends on the size and configuration of the assumed crack, a. Therefore, ΔK may be represented as $\Delta K = \Delta\sigma \cdot g(a)$ where for an internal crack $g(a)$ is defined by $\sqrt{a \times \pi/Q}$ and for a surface crack $g(a)$ is equal to $\sqrt{1.21 \times a \times \pi/Q}$. The crack shape parameter Q takes into consideration the influence of the actual crack shape on the cyclic stress intensity range. The



SPECIMENS CT1-2, WOL 1T-3T / ROOMTEMP (AIR) / f = 10-160 HZ

FATIGUE CRACK GROWTH RATE OF 2-3,5 % Ni-STEELS.



FRACTURE TOUGHNESS K_{Ic} OF SEVERAL STEELS FOR LP-TURBINE SHAFTS AND -DISCS AND FOR GENERATOR-ROTORS.

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SOURCE: Forgings from Gigantic Ingot with 140" Diameter and 881,000 lbs. Weight Part 2

Fatigue Crack Growth Rate and Fracture Toughness of 2 to 3.5% Ni Steels

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MA 4.15a

stress range is calculated for the start-up and shutdown loading conditions of the disk in question.

In order to predict the critical crack depth, a_{cr}, the fracture toughness, K_{IC}, of the disk material must be used in the calculation. In the case of a long-shallow elliptical crack inside of the disk, the critical crack depth is given by:

$$a_{cr} = \frac{K^2_{IC} \cdot Q}{\pi \, \sigma^2_{t2}} \quad (\text{mm})$$

where:

K_{IC} = fracture toughness of rotor material

σ_{t2} = maximum tangential stress at bore surface

The maximum stress, σ_{t2}, which is used for the calculation of the critical crack depth as described above, is derived from peak load service stress and internal residual stress of the disk forging.

Due to the high quality manufacture and quality assurance of the disks, it can generally be assumed that the disk bore region is free of flaws. However, the safety analysis assumes during turbine operation a possible formation of a long shallow semi-elliptical crack with a/2c = 0.2 intersecting the surface of the small locking pin holes as shown in Figure MA 4.17 on page 15 and also Figure NM 4.02, page 28. The critical flaw size for this case is calculated by:

$$a_{cr} = \frac{1}{1.21} \; \frac{K^2_{IC} \, Q}{\pi \, \sigma^2_{t3}}$$

The value of σ_{t3} is based on the stress pattern around the locking pin holes which is influenced by plastic deformation during the first (factory) overspeed test of the rotor. The plastic deformation results in a compressive stress which reduces the value of σ_{t3} below the calculated elastic stress value as illustrated in Figure MA 4.17, page 15.

The safety analysis starts by assuming the critical flaw size. Calculating backwards with this critical flow size must result in an initial flaw size of ≥5 mm equivalent diameter. For this calculation about 4400 start-up cycles are used. This is conservative because a very long operating time would be required to produce such a crack.

The fracture toughness data used in this calculation is based upon extensive tests as discussed in References (5) and (6). With tests performed in accordance with ASTM E-399, fracture toughness K_{IC} is measured as a function of temperature. A test data example is shown in Figure MA 4.15a, page 17. For the safety analysis of disks and shafts, the lower boundary curves of the K_{IC} temperature data scatterbands are used in the calculations.

For correlation purposes, additional tests are performed to determine the fracture appearance transition temperature (FATT), and the nil-ductility transition temperature (NDTT) per ASTM E-208. On this basis, the NDTT in the most critical region of the disk (the bore) is specified at ≤-58°F or -76°F (depending on disk material) which is far below the minimum operational temperature of the rotor.

It is well known that inhomogeneities such as segregation stringers exist in forgings, and extensive work has been done to study this subject on turbine and generator rotors up to the largest monobloc rotor forgings ever produced, as described in References (5), (6) and (7). The studies include microfracto-graphic examinations in the zone of maximum inhomogeneity which occurs between about one-quarter to one-half radius of the rough machined rotor forging. Research to date indicates that the unavoidable minor allow segregations and microporosities in modern forgings have no affect on the mechanical properties important for the safety analysis, fatigue crack growth rate and fracture toughness.

3.3 Manufacture and Quality Assurance

Each LP disk is subjected to a comprehensive and coordinated program of design, manufacture and quality assurance to ensure its reliability throughout the life of the turbine-generator unit. The basic principles, materials, research and development and other aspects of this subject are discussed in detail in References (4) through (7). Following is a brief summary of the key points applicable to LP turbine disks of the type described herein:

The disk forgings are produced from vacuum-degassed alloy steel and are heat treated for an optimum combination of high fracture toughness (throughout the disk volume) and calibrated compressive residual stresses at the hub bore surface. Each disk forging is examined by ultrasonic testing as follows:

- In the rough as-forged condition before heat treatment
- Premachined with contours prior to heat treatment for mechanical properties
- After heat-treatment for mechanical properties

The ultrasonic testing equipment and techniques are in accordance with DIN Standards 54 120 and 54 122, and can detect and measure flaws as small as one mm (0.04 inch) in equivalent diameter.

The evaluation of ultrasonic inspection is based upon reported indications as follows:

1. Isolated single indications of an equivalent defect size of 0.2 inches (5 mm), as per DGS diagram, and larger.
2. All isolated indications causing a decrease of more than 10% of the back reflection.
3. All indications of the linear type or area type, as well as clustered indications, regardless of the size of the single indications in the cluster area.
4. All indications of defects located with a 2 inch (50 mm), zone surrounding the axial center bore.

As indicated in Figure QA 4.01b, material samples are taken from each LP- turbine disk forging near the hub bore surface. The purpose of these samples is to get representative test material for the determination of the nil-ductility transition temperature (NDTT) as a fracture toughness criterion and of the 0.2% offset yield strength as a strength criterion. The results of these mechanical tests in combination with the UT and residual stress measurement results are decisive for the acceptance of the forging.

Before machining out the hub, the following test results of the fully

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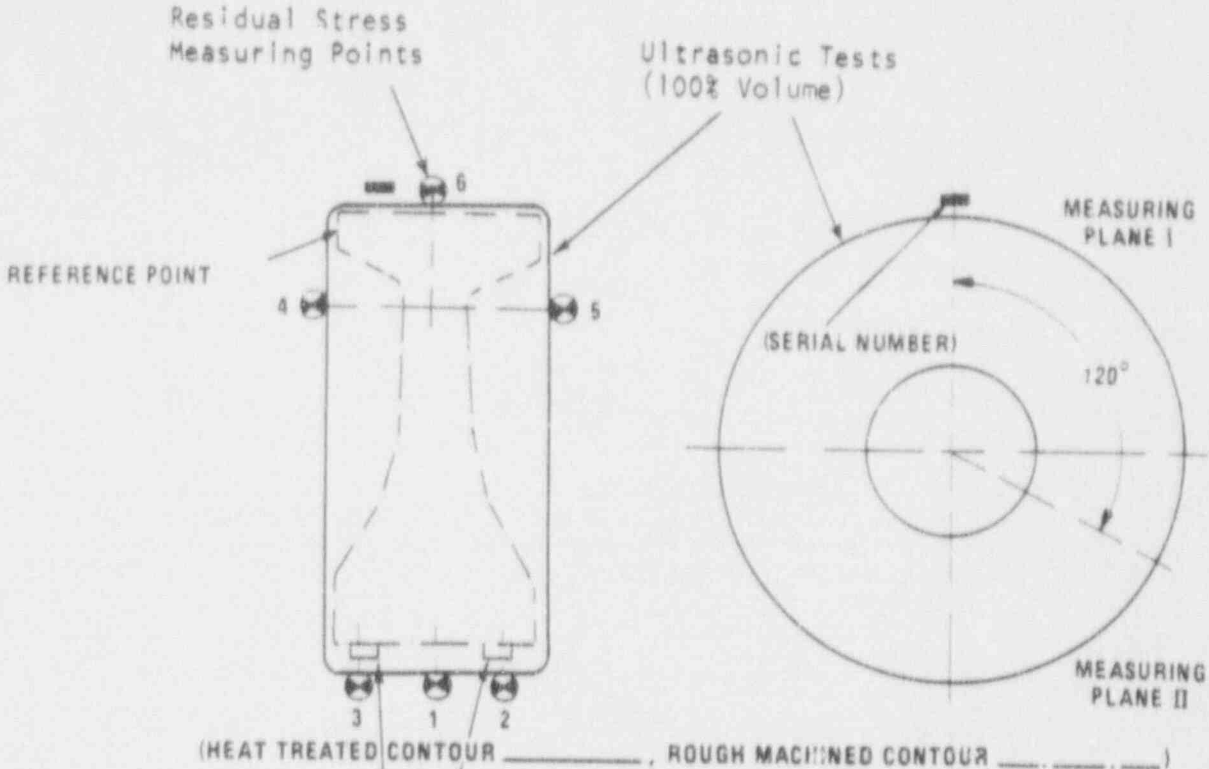
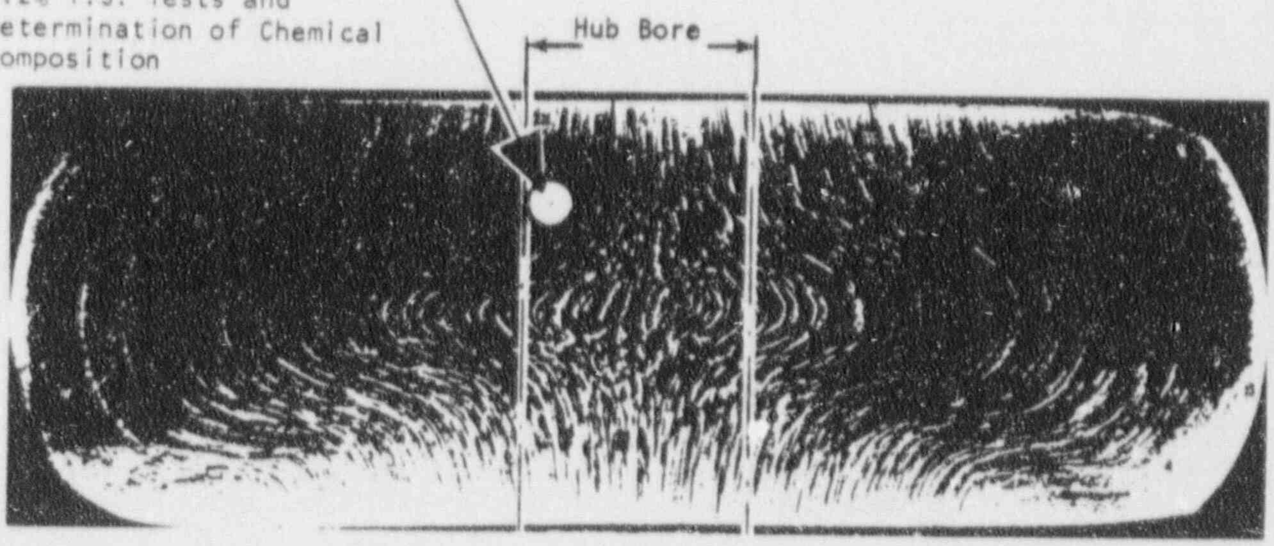


(Sample Position, see Order Drawing)

Material Sample for NDTT and 0.2% Y.S. Tests and Determination of Chemical Composition



Etched Cross Section Through a Small Disk Forging

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Material Testing of LP Turbine Disks

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QA 4.01b

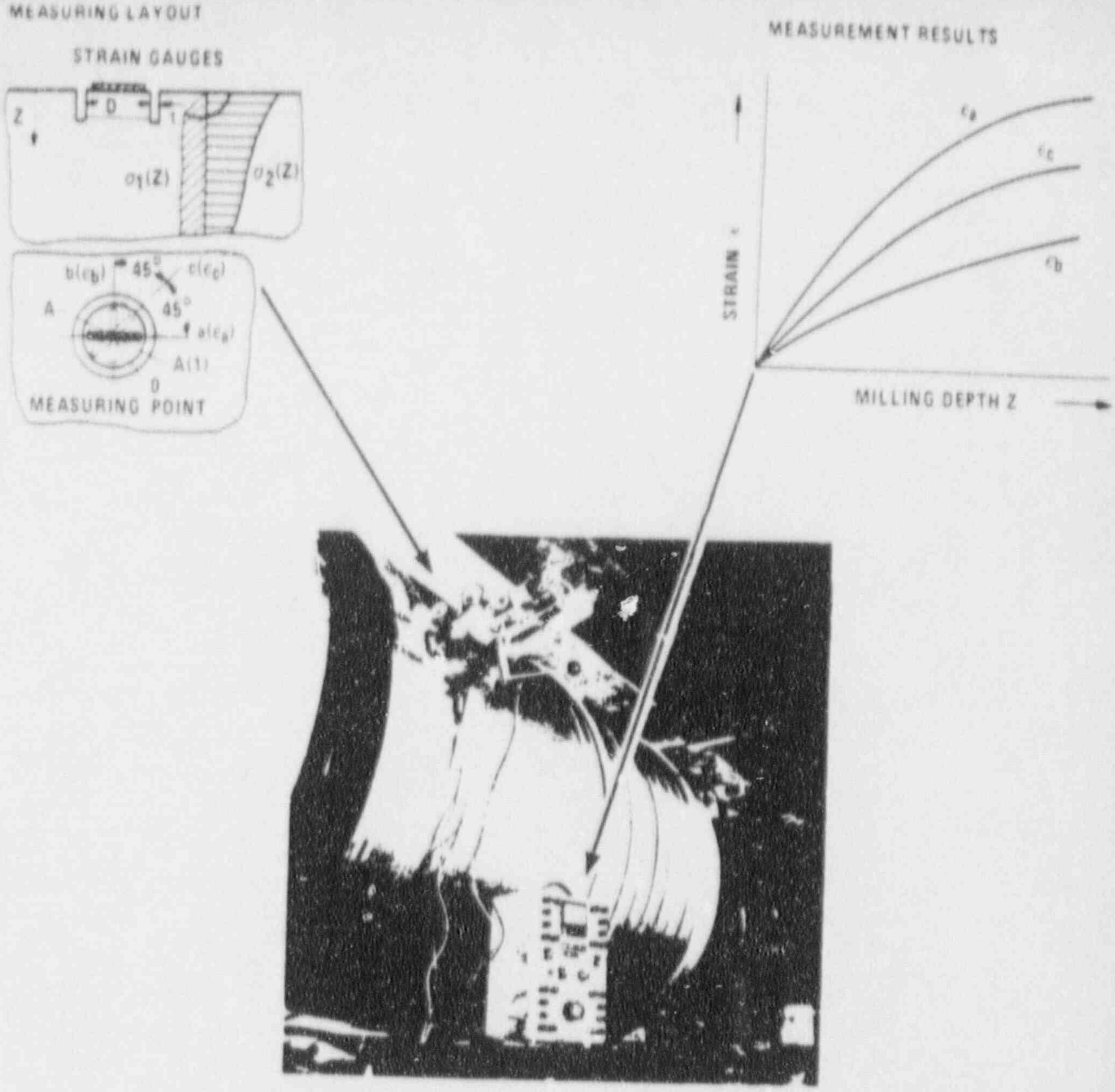
heat-treated forging are obtained and scrutinized:

1. Results of ultrasonic test, covering 100% of the disk volume, and with documentation of all indications as required by the appropriate material purchasing specification.
2. Actual chemical composition of the forging material at locations illustrated in Figure QA 4.01b on page 22.
3. Tensile and drop-weight (NDTT) test results at location indicated in Figure QA 4.01b on page 22.

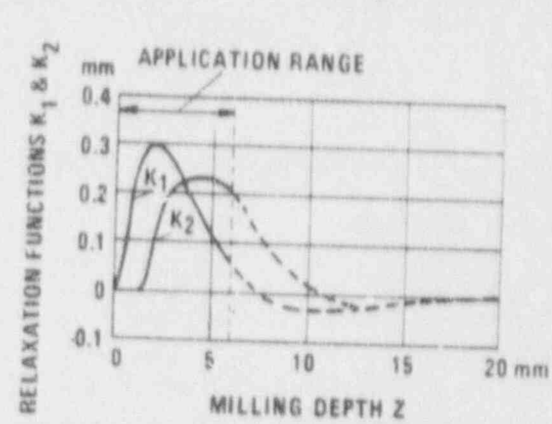
After machining to the dimensions of the order drawing, the bore of the forging is magnetic particle-tested. In addition, residual stresses are measured by the Ring Core Method described in Reference (4) and illustrated by Figures QA 4.02 and QA 4.03. By careful control of heat treatment, desirable residual stress characteristics can be built into the disks. These characteristics are verified by measurements at point 1 to 6 in Figure QA 4.01b on page 22 before and after machining for the first of a batch of similar disk forgings. Subsequent disks of the same batch are checked only at point 1.

3.4 LP Rotor Assembly

Figure NM 4.01 shows the assembly line for low-speed LP turbine rotors which includes shrink fitting the disks and couplings on the shaft, as well as inserting the blades into the grooves of the shrunk on disks. To date, more than 200 disks have been shrunk on LP turbine rotors using the tecnniques described herein.



RELAXATION FUNCTIONS K_1; K_2 (D = 14 mm, t = 2 mm)



BASIC EQUATIONS FOR STRAINS σ_a; σ_b, σ_c

$\sigma_{a,b,c} = f_1 (\epsilon_{a,b,c}, Z, K_1, K_2, E, \mu)$

WHERE: $K_1, K_2 = f_2$ (Z, D, t, STRAIN GAUGE GEOMETRY)

E = MODULUS OF ELASTICITY

μ = POISSON'S RATIO

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Ring Core Method for Determination of Residual Stresses

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QA 4.02

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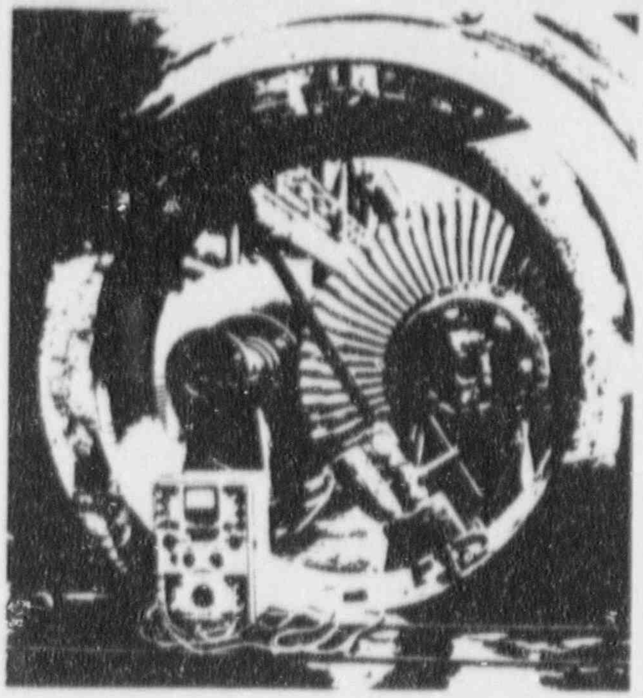
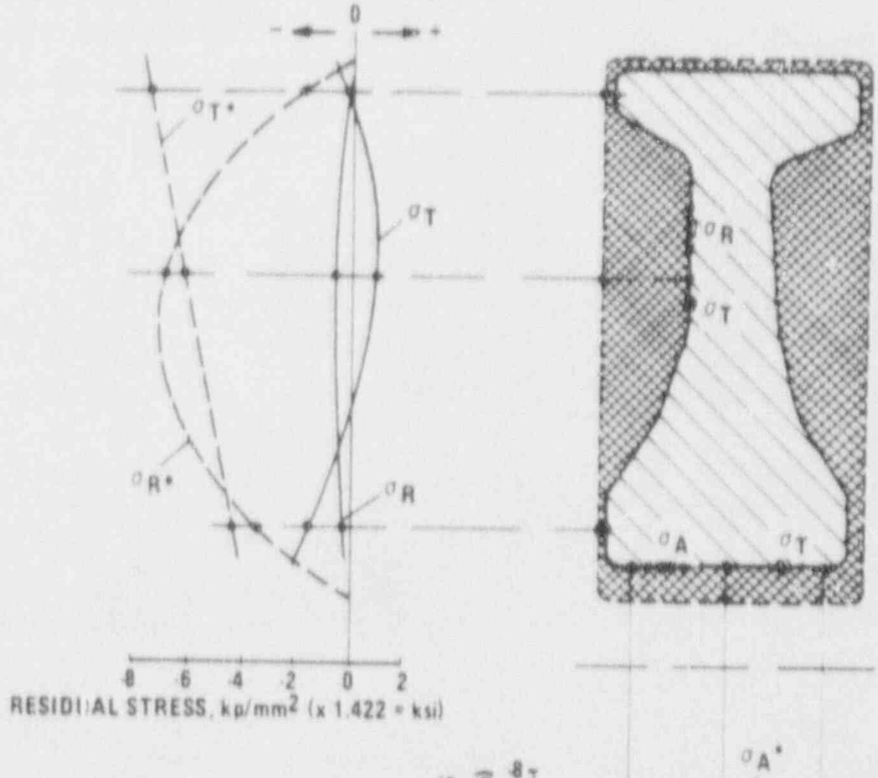
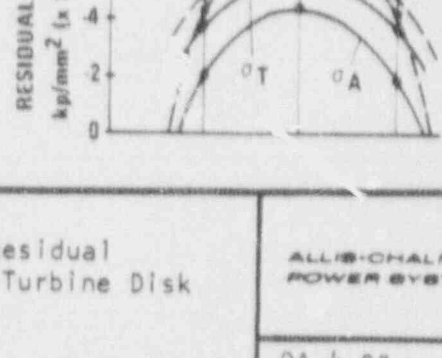




σ* RESIDUAL STRESS BEFORE MACHINING (TURBINE DISK PROFILE — — — —)

σ RESIDUAL STRESS AFTER MACHINING (TURBINE DISK PROFILE ————)



Determination of Residual Stresses in an LP Turbine Disk

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QA 4.03

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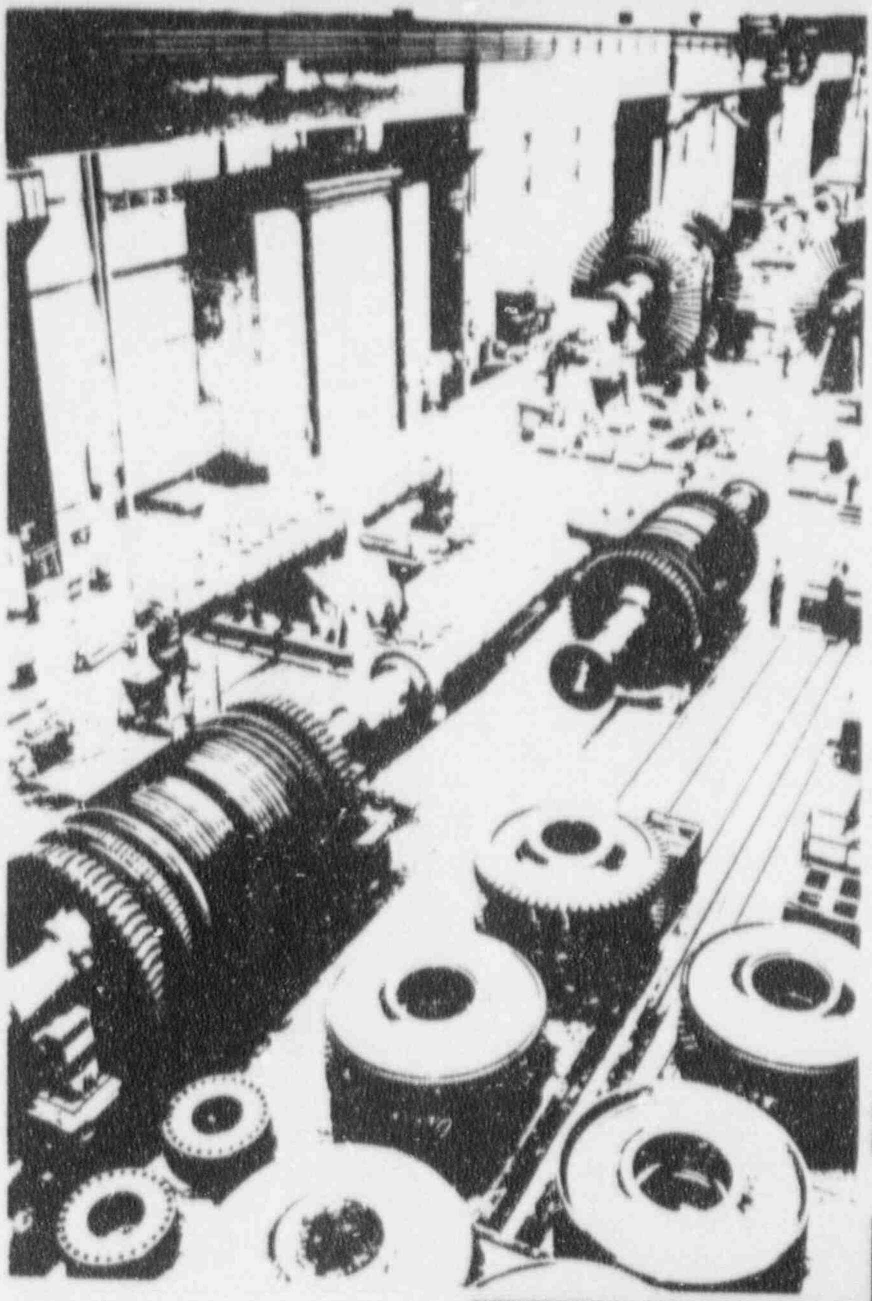
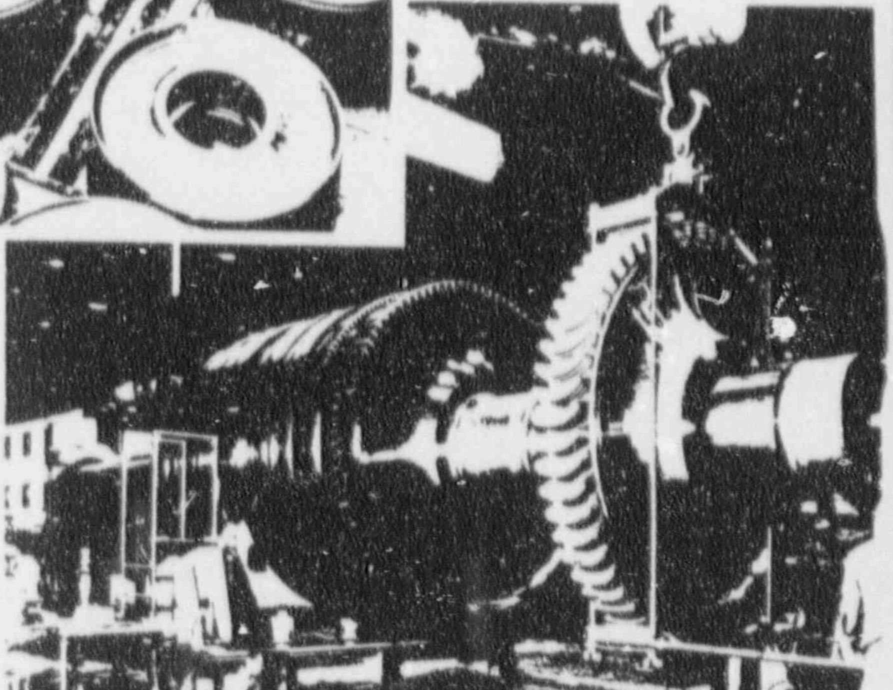




Production of LP Turbine
Rotors with Shrunk-on Disks

ALLIS-CHALMERS POWER SYSTEMS, INC.

NM 4.01

The shrinking process is performed with the shaft in the horizontal position, and with rotation of the rotor during the process, which provides the following benefits:

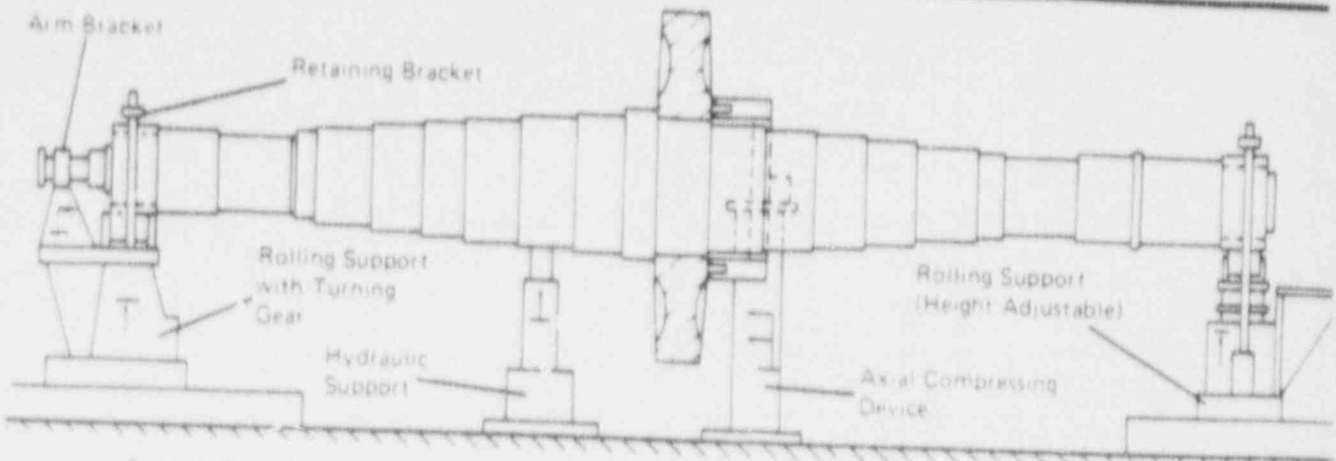
1. Defined heat transfer from disk to rotor, even during the initial stage of the shrinking process when the clearance is still greater than zero because the disk always rests with its weight on the shaft.

2. Exact positioning of the disk by the axial compressing device during the initial stage of the shrinking process when the clearance is still greater than zero and turning gear is in operation.

3. Easy correction of rotor run-out by stopping the turning gear with the shaft in the correct position during the initial stage of the shrinking process for a calculable period of time.

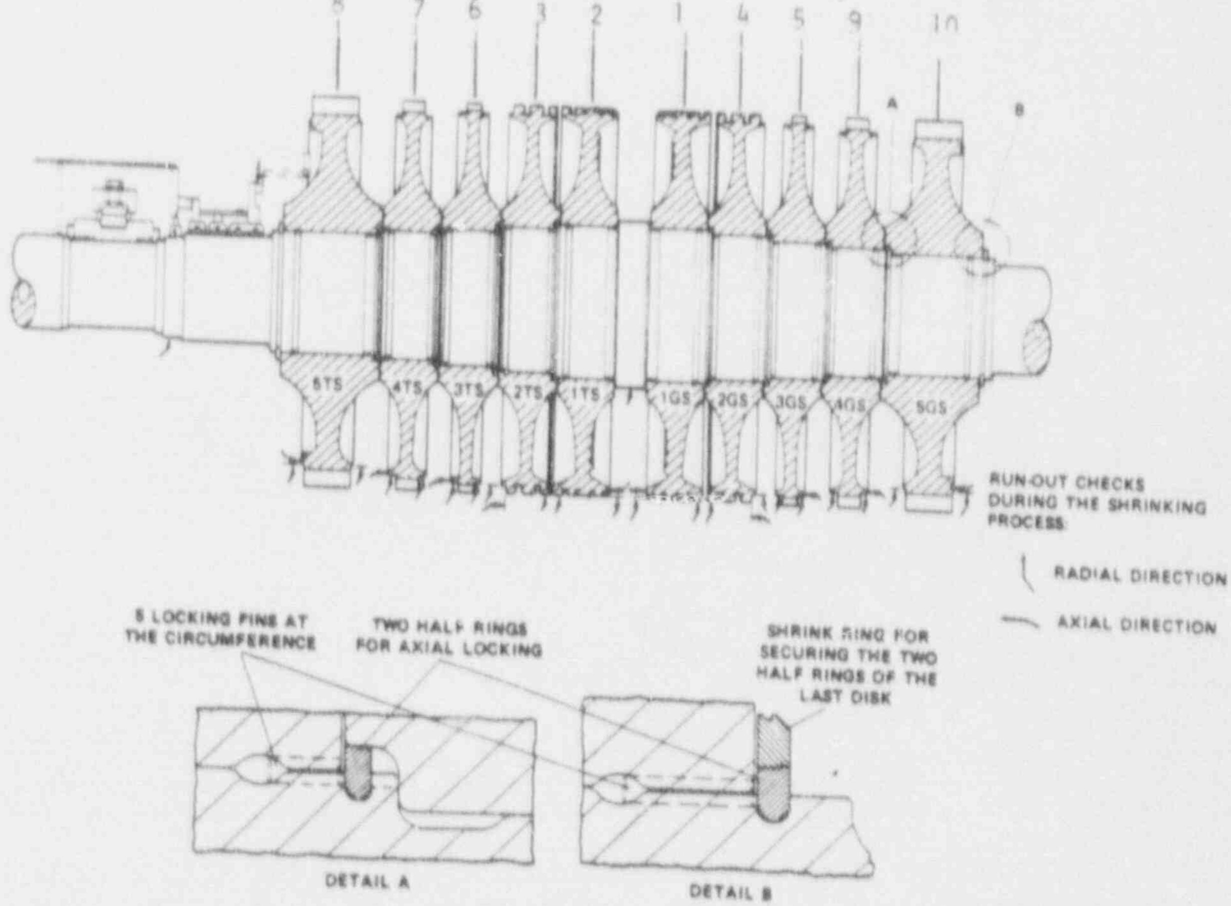
4. Axial and radial run-out checks during the entire shrinking process with shaft in its operating position.

A shrinking stand and heating oven are the main facilities required for shrinking disks on shafts. The shrinking stand (Figure NM 4.02) has two rolling supports, one with a turning gear and the other with height adjustment. Before turning the rotor, the height of the second support has to be adjusted in such a way that the rotor rests on the arm bracket, which is mounted either on the left or the right rolling support, depending on whether disks are to be shrunk on the generator or turbine end of the shaft. A hydraulic support device is used to prop the rotor



Stand for Shrinking-on Disks with Horizontal Rotating Process

Typical Sequence of Shrinking



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Shrinking of Disks on LP Turbine Shaft	ALLIS-CHALMERS POWER SYSTEMS INC
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while one of the rolling supports is removed to allow a hot disk to be slipped over the shaft end. A retaining bracket at the other rolling support prevents the shaft from tilting. A compressing device pushes each disk against the appropriate axial positioning shoulder of the shaft while the disk cools down to ensure a shrink fit at the correct axial position.

The oven is of the hood type with an electrical heating system which transfers heat uniformly and accurately controls the disk temperature. Before being put into the oven, the disk is mounted on a loading rack and leveled so that it lies in an exac-ly horizontal plane. After heating to about 700 F, it is picked up by the crane, moved to the shrinking stand and slipped over free end of the shaft. Then, the rolling support is remounted at the free end, and the hydraulic support at the middle of the shaft is withdrawn. Next, the disk is moved into its final position and pressed against the appropriate axial shoulder by the compressing device.

The shrinking process really starts when the disk is in contact with the shaft and the crane hook is disconnected. A large heat transfer takes place between the hot disk and the cold shaft at the top of the shaft. The shaft is turned alternately in both directions to provide a uniform shrinking process, checked by run-out measurements. After about 45 minutes, the entire shrink-fit becomes zero, -- indicated by equal run-out results for shaft and disk. While the disk cools

down to about 120 F, run-out checks are performed with the shaft being rotated at about 1.5 rpm by the turning gear. This procedure for shrinking on a disk takes about one day.

After the disk and shaft have cooled off, five short locking pins are installed at one disk side between disk and shaft, and two half rings are assembled to axially lock the disk itself and the five pins (see Figure NM 4.02, page 28). The shrink fit, which results from the disk bore being about 0.1 inch smaller in diameter than the shaft, is sufficient to hold the disk firmly in position at speeds in excess of 120% of rated speed. The five locking pins are only an additional protection against circumferential displacement of the disk. The two half rings for axial locking are caulked into the shaft groove so they cannot drop out while the next disk is being moved into place where it holds them in position against centrifugal force. The two half rings for the last disk are secured by a locking ring which is shrunk over the half rings. A stop in the outer diameter of the half rings holds this locking ring in position even if there is no shrink force.

Numerous run-out checks are made to ensure that all disks are properly shrunk on the shaft as indicated in Figure NM 4.02, page 28. The maximum permissible radial run-out at any disk circumference is about 3 mils which is about the same as the allowable run-out of the shaft at its center. Since the run-out check in axial direction determines the correct fit of the disks against the axial shaft shoulders, they are required to be twice as accurate, namely within 1.5 mils.

3.5 Balance and Overspeed Test

Before delivery, each completed bladed LP rotor is balanced, and subjected to an overspeed test at 120% of rated speed for two minutes at a minimum temperature of 59°F (see Figure TT 4.01). All other turbine and generator rotors are also subjected to a 120% overspeed test.

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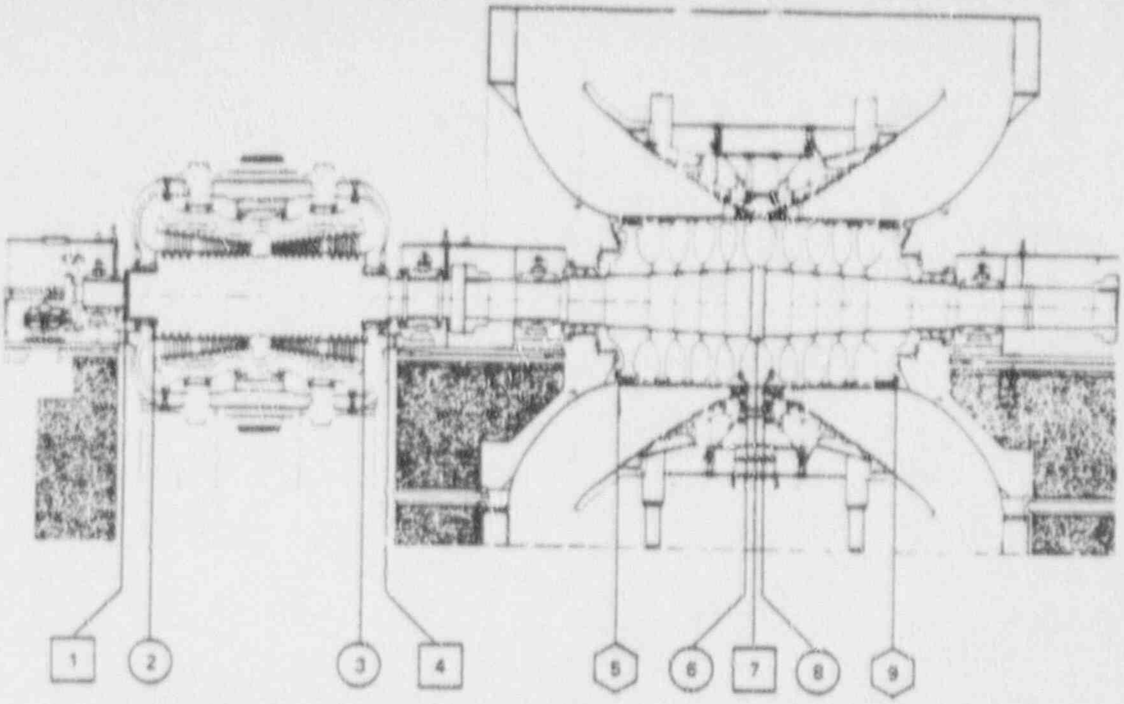
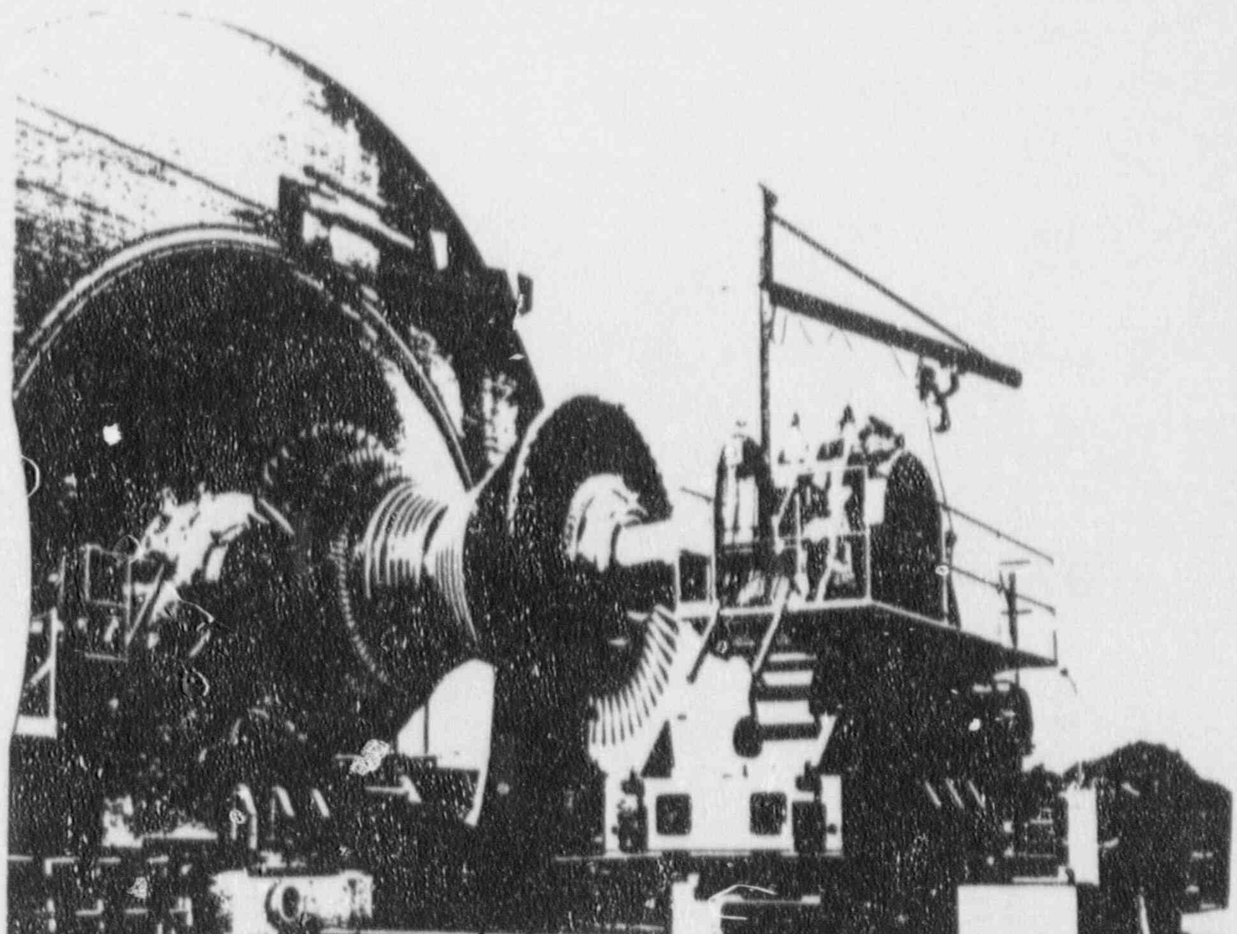


Balancing Planes of an 1800 rpm Turbine for LWR Applications

FACTORY   POWER PLANT   FACTORY & POWER PLANT



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Balancing and Overspeed Testing of Turbine Rotors	ALLIS-CHALMERS POWER SYSTEMS, INC.
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4. PROBABILITY OF FAILURE DUE TO OVERSPEED

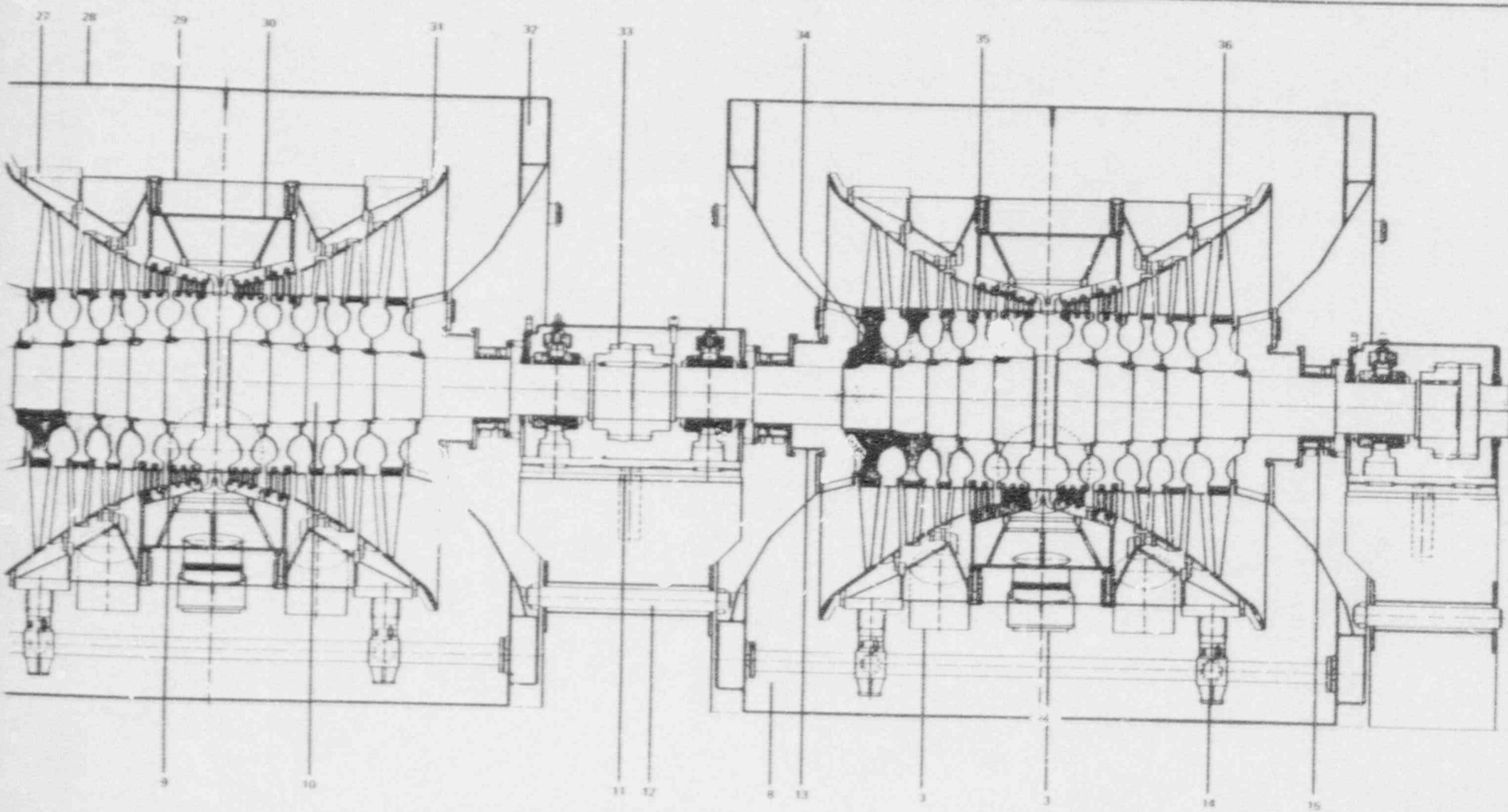
4.1 Introduction

This section contains an analysis and calculation of the probability of a >120% speed event. Although such an event would not necessarily result in an LP rotor failure, it is conservatively assumed for purposes of this analysis that the failure probability is equal to the overspeed probability.

The design and operation of the overspeed prevention features of an 1800 r/min turbine-generator for Light Water Reactor (LWR) applications are described in A-CPSI's previously issued report #E415 (8). The reader is referred to report #E415 for basic descriptive information which will not be repeated in this report. However, for convenient reference, a copy of the speed control system schematic diagram NC3.06b is included in this report.

The methodology of the failure probability analysis is based on IEEE Std. 352-1972 (ANSI-N41.4) entitled "General Principles of Reliability Analysis of Nuclear Power Generating Station Protection Systems." (9) The study consisted of the following basic steps:

- Analysis and preparation of the reliability block diagram for a >120% speed event.
- Establishment of random failure rates for individual elements of the system.
- Analysis and preparation of fault trees leading to a >120% speed event.



9 HP TURB E OUTER CASING
HP TURBI E INNER CASING
1 INNER CASING CENTERING BOLT
2 REMOVABLE HP STEAM SEALS
3 COMBINED THRUST AND JOURNAL BEARING
4 SHAFT VIBRATION SENS R
5 BEARING SHELL VIBRAT ON SENSOR
6 LP TURBINE JOURNAL BEARING
7 MISSILE CRASH RING

28 LP TURBINE OUTER CASING
29 LP TURBINE INNER CASING OUTER SHELL
30 LP TURBINE INNER CASING INNER SHELL
31 LP TURBINE DIFFUSER
32 OUTER CASING REINFORCING RING
33 LP TURBIN PLING
34 SHRUNK O ROTOR DISC
35 INNER SH NTERING BOLT
36 FREE STAN ING LP URBINE END BLADES

LONGITUDINAL SECTION OF
1800 RPM FOUR FLOW
NUCLEAR TURBINE

ALLIS-CHALMERS
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- Preparation of a computer program and calculation of the >120% speed probability.
- Qualitative analysis of common mode failure possibilities.
- Review and preparation of conclusions and recommendations.

Throughout the study conservative assumptions and methods were used consistent with safety analysis such that the final result may be considered a safe estimate of the probability of >120% speed failure of an LP rotor.

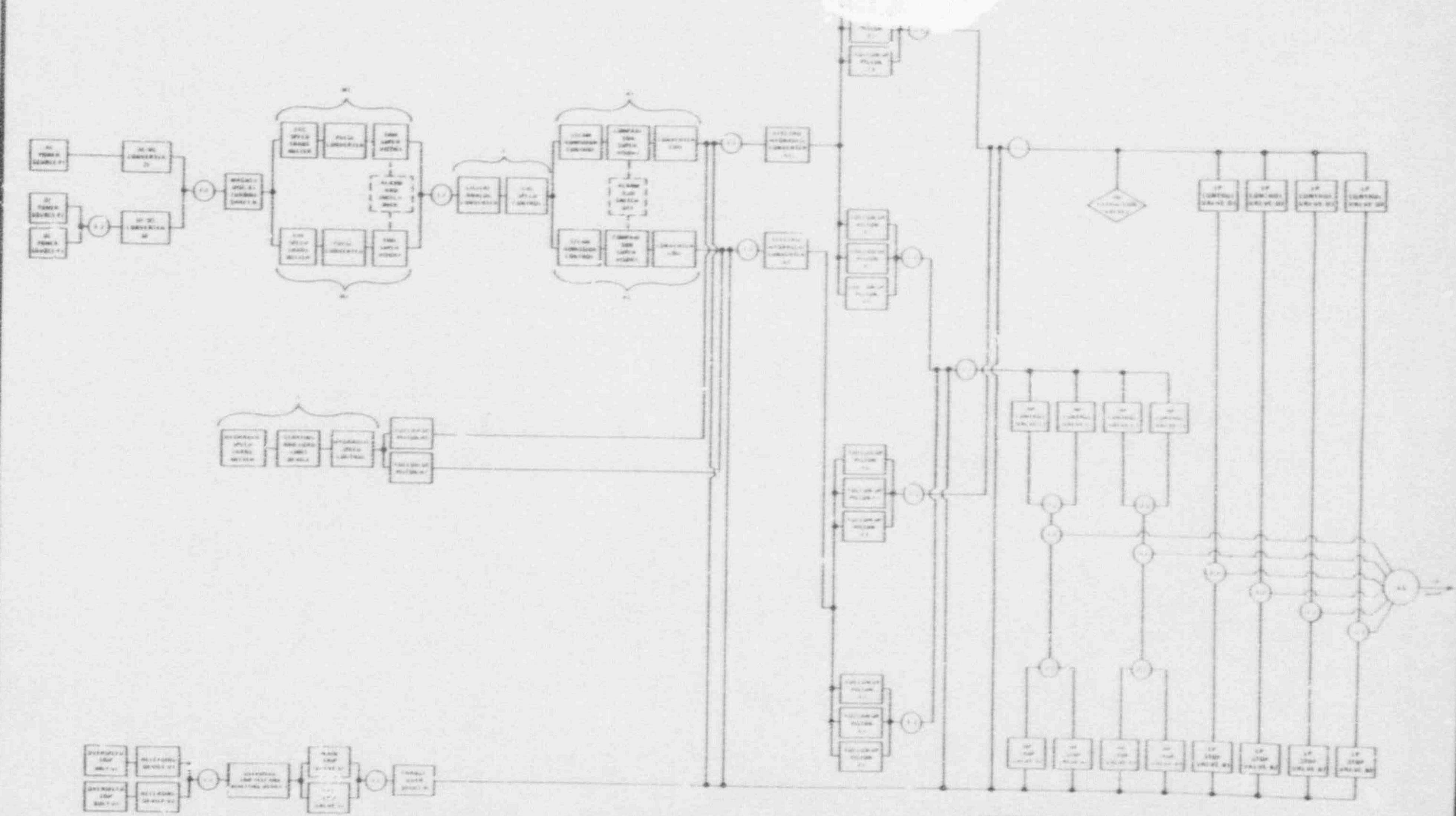
4.2 Reliability Block Diagram

The Reliability Block Diagram NC 5.01a shows the overspeed prevention system of a 4-flow turbine including the Electro-Hydraulic Control (EHC), the Mechanical Hydraulic Speed Control (MHC), and the Overspeed Trip. Only one of these three systems is required to prevent a >120% speed event. All three systems form a one-out-of-three* signal to the control valves. Each system has built-in redundancies of certain components contributing to high reliability.

4.2.1 Electro-Hydraulic Control

The EHC receives power from one a.c. power source, and either one or

* As used in this report, the term one-out-of-three means proper operation of any one of three system elements is sufficient to prevent a >120% speed incident, and similarly for all terms x-out-of-y.



RELIABILITY BLOCK DIAGRAM 120%
SPEED CONTROL SYSTEM OF A
4 FLOW REHEAT TURBINE
GENERATOR FOR LIGHT WATER

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two independent d.c. power sources depending on the reliability and characteristics of the primary a.c. source. The internal EHC power supply is formed by a one-out-of-two supply from the a.c./d.c. converter and the d.c./d.c. converter via diodes to provide a continuous power supply for the EHC.

The speed measuring device is a two channel system with automatic detection and alarm for failure of the primary channel and automatic switch-over to the back-up channel. The steam admission control with electro-hydraulic converters is built as a two channel system with an internal supervisory subsystem. If a channel fails, it will be switched off and an alarm given while operation continues on the remaining channel. In regard to overspeed prevention, the follow-up pistons which form the control fluid pressure signal for the control valves have a redundancy of one-out-of-three taken twice.

4.2.2 Mechanical-Hydraulic Control

The MHC is a one channel system positioning two independent follow-up pistons to form two separate control fluid pressure signals for the two EH converters.

4.2.3 Overspeed Trip System

The overspeed trip signal for the stop and control valves is initiated by two independent trip bolt and releasing devices in a one-out-of-two system. Also, the proper function of only one of the two main trip valves is sufficient to close the stop and control valves. The over-

speed trip test and resetting device and the changeover device operate only during test of the trip system. During normal operation and a real trip of the turbine, these devices are inactive in their safe position.

4.2.4 Stop and Control Valves

To prevent overspeed, the control valves and extraction valves receive a one-out-of-three closing signal from the EHC, MHC and trip system. Four HP control valves are arranged one in each of four HP admission pipes to the turbine. An independent HP stop valve is located immediately upstream of each HP control valve. The four HP stop valves and four HP control valves are in series forming a one-out-of-two system in each of the four admission lines. No interconnection between the four separate HP stop valves and control valves is required for throttle controlled operation with full-arc admission of the HP turbine. However, two interconnections of a pair of two HP stop and control valves is established by the heating steam supply pipes to the two MSRs as shown in Fig. E 5.119. The valve arrangement without any interconnection would be the most reliable system because one stop and control valve in one admission line has to fail to drive the turbine-generator to a >120% speed in a very short time. Having an interconnection, especially between all four valves, would lead to a higher overspeed probability because a failure of one of the stop valves and one of the control valves which are interconnected produces a >120% speed event. For the failure probability, the conservative approach was taken, that the heating steam shut-off valves between the two pairs of stop and control valves are fully open during a load rejection or trip, despite the fact that they get a closing signal in case of a unit trip.

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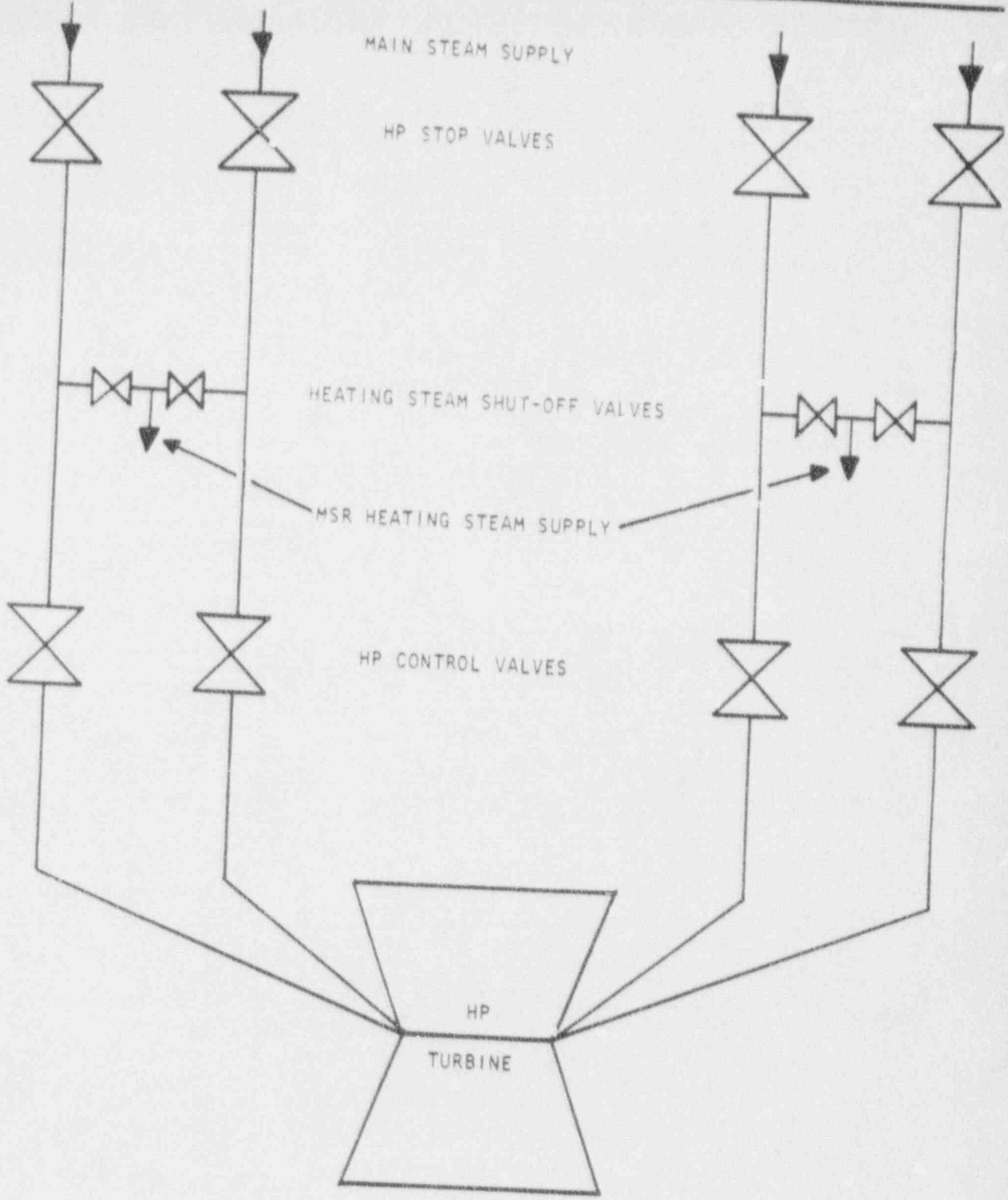

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HP STOP AND CONTROL
VALVE ARRANGEMENT
LWR APPLICATIONS

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E 5.119

Each admission pipe to an LP turbine is equipped with a butterfly-type stop valve and a butterfly-type control valve. A closing of only one of the two valves in each line is sufficient to avoid an overspeed event because there is no inter-connection between these admission

The Schematic Reliability Block Diagram E5.120 shows the same overspeed prevention system as in NC5.01a, page 36, except in terms of symbols for the system elements from A to V and the component numbers 1 to 36 which are used in the computer program to calculate failure probability. This diagram also gives a visual impression of the very high redundancy of the signal to the control valves, and the importance of a high reliability of the stop and control valves and the trip system in regard to overspeed protection.

4.3 Failure Rates

The following is a summary of analysis of the system components and elements which was done to define the failure rates of each element. In general, the approach was to calculate the failure rates based on actual operating experience using a statistical confidence level of 95%. For example, operating experience with stop valves similar to current design totals 5.3×10^7 valve-hours; and during this operation there were two failures which could lead to a >120% speed incident. For a confidence level of 95% and two observed failures, the Thorndike chart (see Figure NC4.12) indicates the failure rate should be calculated by assuming 6.25 failures; therefore, the failure rate, $\lambda = 6.25/5.3 \times 10^7 = 0.117$ failure per million hours.

SYMBOL	NAME	COMPONENT NUMBER
A1 4	HP STOP VALVES 1 TO 4	23-26
B1 4	LP STOP VALVES 1 TO 4	33-36
C1 4	HP CONTROL VALVES 1 TO 4	19-22
D1 4	LP CONTROL VALVES 1 TO 4	27-30
E1 3	FOLLOW UP PISTONS 1 3 BATTERY I HP	15
E4 6	FOLLOW UP PISTONS 4-6 BATTERY I LP	16
F1 3	FOLLOW UP PISTONS 1 3 BATTERY II HP	13
F4 6	FOLLOW UP PISTONS 4-6 BATTERY II LP	14
G1 G2	ELECTRO HYDRAULIC CONVERTER 1 AND 2	11 12
H1 H2	MHC FOLLOW UP PISTON 1 AND 2	5 6
I	MECHANIC HYDRAULIC CONTROL (COMPLETE)	2
K1 K2	ADMISSION CONTROL 1 AND 2 (COMPLETE)	7 8
L	EHC SPEED CONTROL (COMPLETE)	
M1 M2	EHC SPEED MEASURING DEVICE (COMPLETE) 1 AND 2	4
N	MAGNET DISC AT TURBINE SHAFT	
O1 2	AC DC AND DC DC CONVERTER 1 AND 2	
P1 3	POWER SOURCES 1 TO 3	3
R	CHANGE OVER DEVICE	
S1 S2	MAIN TRIP VALVE 1 AND 2	
T	OVERSPEED TRIP TEST AND RESETTING DEVICE	
U1 U2	OVERSPEED TRIP BOLT 1 AND 2	1
V1 V2	RELEASING DEVICE 1 AND 2	

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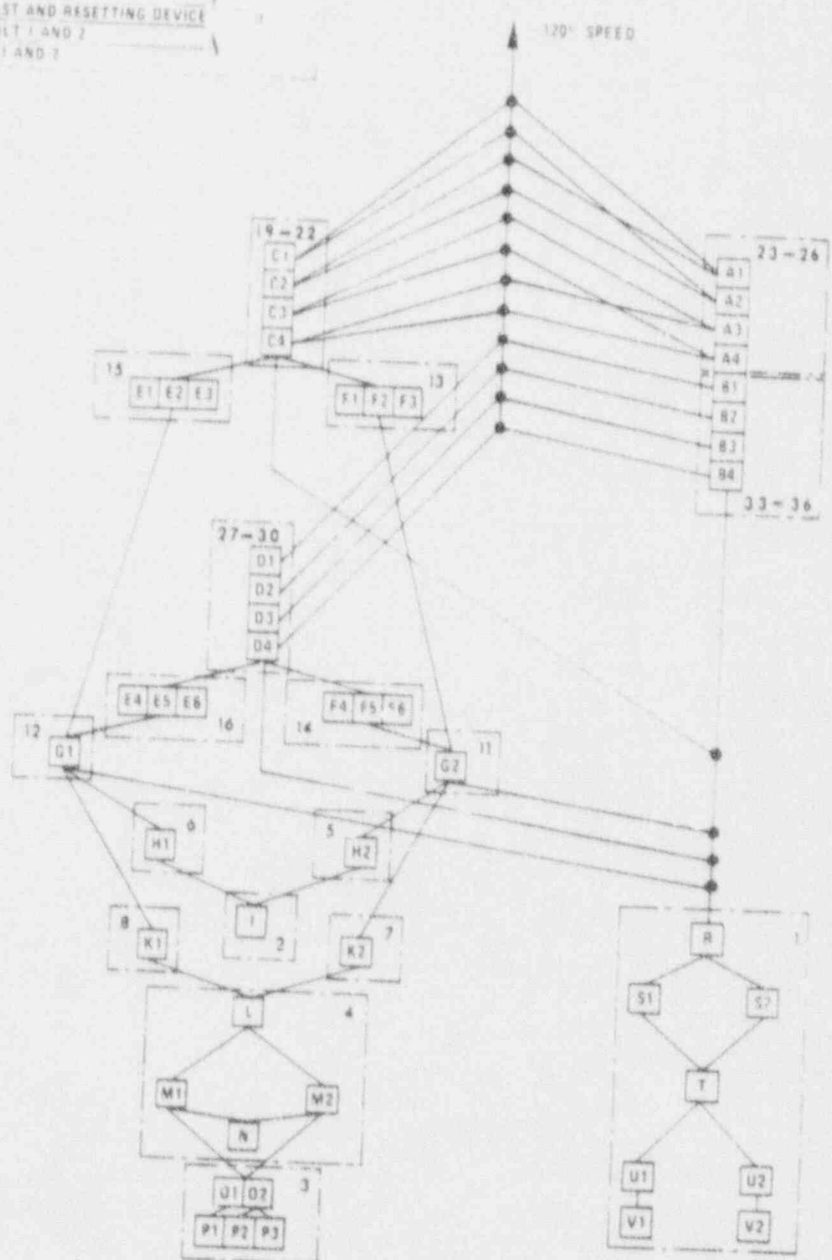

SCHEMATIC RELIABILITY BLOCK DIAGRAM WITH COMPONENT NUMBERS 4-FLOW TURBINE FOR LWR APPLICATIONS

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E5.120

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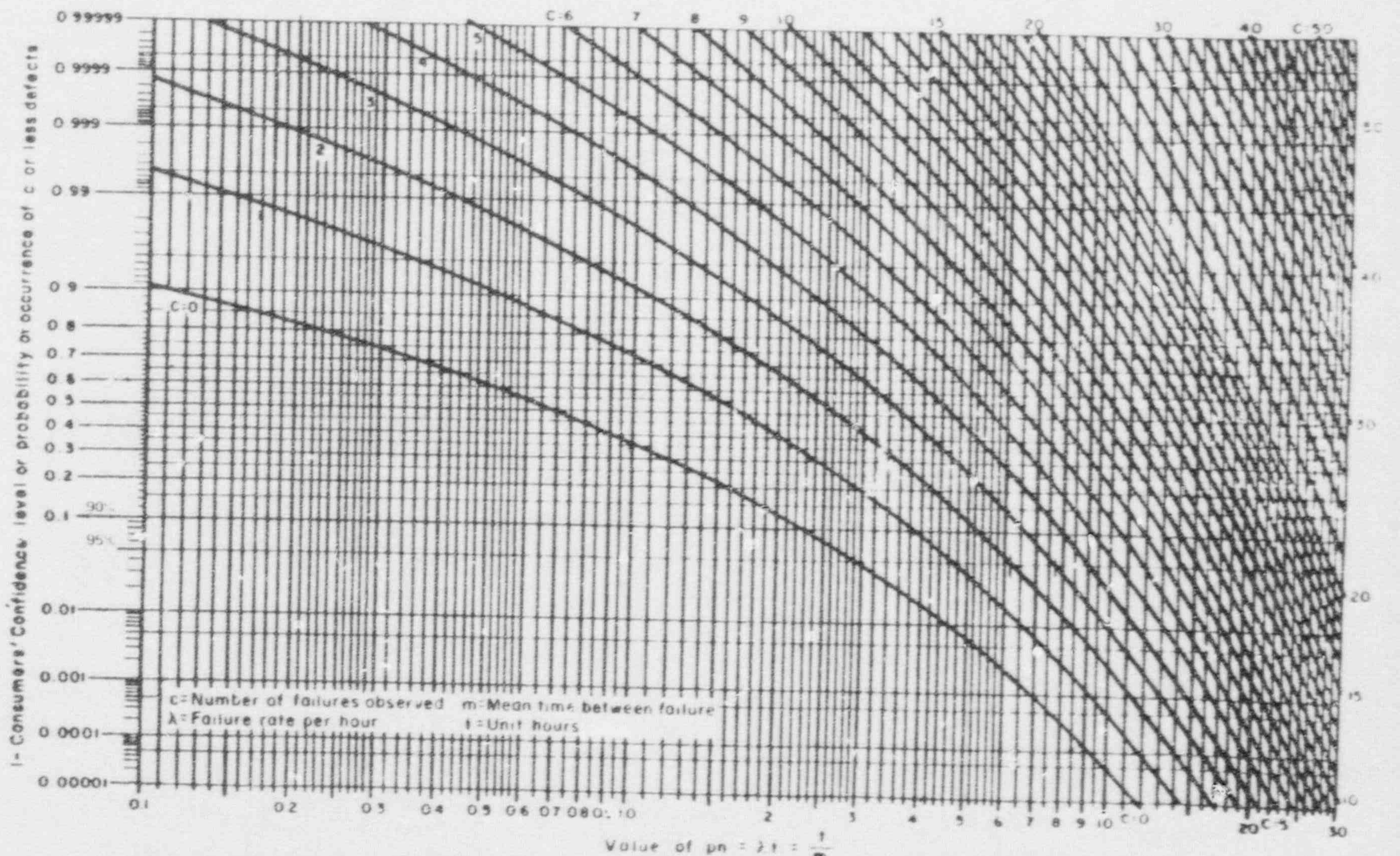
Relationship Between

1. Probability of acceptance and expected defects in a sample (units or unit hours) for various observed failures in the sample
2. 1-Consumers' Confidence and expected defects in a sample (units or unit hours) for various observed failures in the sample

Cumulative curves for poisson distribution (Thorndike chart)



For determining probability of occurrence of c or less defects in a sample of n pieces selected from an infinite universe in which the fraction defective is p

THORNDIKE CHART

ALLIS-CHALMERS POWER SYSTEMS INC.

NC 4.12

The values for operating experience and failures given in the overspeed failure rate chart are cummulative from the starting dates shown through January 1, 1975.

The failure rates of the elements also may be considered to include all interconnections from element to element so that the calculated failure probability covers the entire overspeed prevention system.

The chart "Overspeed Failure Rates of the Stop and Control Valves and Turbine Trip and Control System Elements" shows the result of investigation. The chart lists the operating experience, and the number of failures which could lead to a >120% speed event for each element of the system. The applicable experience information and the applicable failures are included in the data chart. The mean time between failure (MTBF) and the failure rate λ are calculated on a 95% confidence level using the Thorndike Chart. With this conservative 95% confidence level, the MTBF and λ for an element which had zero historical failure are calculated with an assumed number of 2.95 failures. For one historical failure, 4.7 failures are assumed, and for an element with two historical failure, the MTBF and λ have been calculated with an assumed number of 6.25 failures.

The mission times "t" are based on a bi-weekly test of the trip system and the stop and control valves, and a yearly inspection (e.g. during reactor refueling) of the entire overspeed prevention system. Experience indicates it can be safely assumed that no significant wear-out of our system elements will occur between yearly inspections so that purely random failure rates are valid.

OVERSPEED FAILURE RATES OF THE STOP AND CONTROL VALVES AND THE TURBINE TRIP AND CONTROL SYSTEM ELEMENTS (1) (7)

Elements	Component Number	Name	Experience Since	No. of Components	Component Hrs.	Failures (2)	MTBF with (3) 95% Confidence	t, Mission (4) Time	λ, Failure Rate with 95% Confidence
A1-4 & B1-4	21 & 22	Stop Valves	1958	884	5.3×10^7	2	8.48×10^6 HRS	336 HRS	$0.117/10^6$ HRS
C1-4 & D1-4	19 & 20	Control Valves	1965	506	1.85×10^7	1	3.93×10^6 HRS	336 HRS	$0.254/10^6$ HPS
E1-6 & F1-6 & H1, H2	5 & 6 & 13 & 14 & 15	Follow-Up Pistons	1954	1360	7.2×10^7	0	24.4×10^6 HRS	8760 HRS	$0.041/10^6$ HRS
G1, G2	11 & 12	Electro-Hydr. Converters without Coils	1963	86	2.65×10^6	0	0.90×10^6 HRS	8760 HRS	$1.11/10^6$ HRS
I	2	Mechanical Hyd. Control	1954	427	1.68×10^7	1	3.57×10^6 HRS	8760 HRS	$0.28/10^6$ HRS
K1, K2	7 & 8	Admission Controls with Coils	1963	86	2.65×10^6	4	0.29×10^6 HRS	8760 HRS	$3.43/10^6$ HRS
L & M & N	4	EHC Speed Control & Speed Measuring Device	1963	86	2.65×10^6	0	0.90×10^6 HRS	8760 HRS	$1.11/10^6$ HRS
O & P	3	Power Supply	1963	86	2.65×10^6	0	0.90×10^6 HRS	8760 HRS	$1.11/10^6$ HRS
R		Change-Over Device	1966	53	1.45×10^6	0		336 HRS	$10/10^6$ HRS (5) $0.0001123/10^6$ HRS
S1, S2		Main Trip Valves	1958	278	1.43×10^7	0	4.85×10^6 HRS	336 HRS	$0.206/10^6$ HRS
T	1	Overspeed Trip Test and Resetting Device	1958	242	1.4×10^7	0		336 HRS and 8760 HRS	$10/10^6$ HRS (6) $0.0335/10^6$ HRS $0.000225/10^6$ HRS
U1, U2		Overspeed Trip Bolts	1958	486	2.78×10^7	7	2.12×10^6 HRS	336 HRS	$0.471/10^6$ HRS
V1, V2		Trip Release Devices	1958	486	2.78×10^7	0	9.42×10^6 HRS	336 HRS	0.1[illegible]$/10^6$ HRS

(1),(2) For predicting random failures of the elements which could lead to a > 20% overspeed event historical failures in "control and/or stop valve opening direction" are listed. This includes "control or stop valve stay open failures," too.

(3) MTBF (Mean Time Between Failures) and λ (Failure Rates) are calculated on a 95% confidence level using the "Cumulative Curves for Poisson Distribution (Thorndike Chart)".

(4) Mission time is defined by the bi-weekly automatic test of the trip system and stop and control valves and a yearly inspection of the overspeed control system.

(5),(6) The failure rates of these elements which are only required during testing are defined as described in this report.

(7) Historical data through January 1, 1975.

The mission time "t" and failure rate "λ" are used to calculate P the probability [10] of failure with the following formula:

$$P = 1 - e^{-\lambda t}$$

This equation can be expanded into an infinite series as follows:

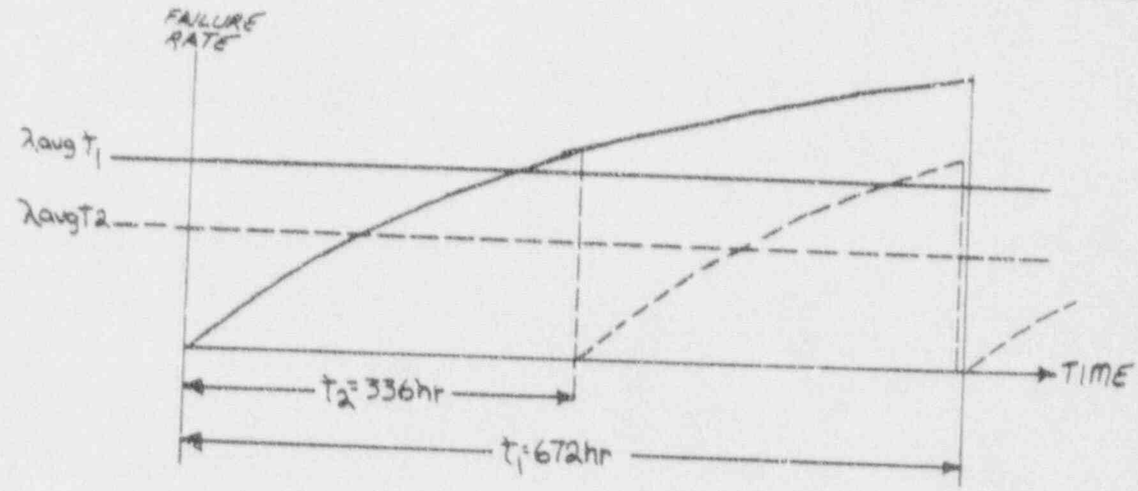
$$P = \lambda t - \frac{(\lambda t)^2}{2!} + \frac{(\lambda t)^3}{3!} - + - \ldots\ldots$$

For $\lambda t \leq 0.3$ the following approximation including the fourth power term is used in the computer program to calculate

$$P = \lambda t - \frac{(\lambda t)^2}{2!} + \frac{(\lambda t)^3}{3!} - \frac{(\lambda t)^4}{4!}$$

The failure probability of all components is assumed constant over the whole life of the unit based upon periodic testing and maintenance which continually checks and repairs or replaces components to maintain the reliability of the system; in other words we are assuming there will be no significant wear-out effects. In this connection it should be noted that in our study of operating experience and past failures going back to 1958, there is no indication of any component wear-out trends or effects.

The historical failure rate of the stop valves for example is based on the periodic testing of these valves which was actually performed in the past, and is an average λ for the actual test interval. Most of the valves in the sample were not equipped with an automatic turbine tester (ATT), therefore, valve testing was not performed bi-weekly (monthly test intervals can be assumed as average). Considering the importance of testing, and having the benefit of the ATT, valve testing of nuclear units is recommended bi-weekly. As the following diagram indicates, we took the conservative approach, and did not change the historical failure rate due to more frequent testing, but rather used the bi-weekly test time as the mission time t of the stop valves.



AVERAGE FAILURE RATES OF REDUNDANT SYSTEMS

The historical $\lambda = \lambda_{avg\ t1}$ based on a four-week test interval. With bi-weekly test interval, λ and t will decrease to $\lambda = \lambda_{avg\ t2}$ and $t = t2$. However, we actually used the larger $\lambda = \lambda_{avg\ t1}$ for the bi-weekly test t2.

Following are brief discussions of the experience, failure and other information used to establish the failure rate of each system element.

4.3.1 HP and IP Stop and Control Valves, Elements A 1-4 to D 1-4

The applicable historical data is as follows:

Stop Valves

Operation experience since	1958
Valves in operation	884
Valve hours	5.3×10^7 hrs
Failures	2

Control Valves

Operation experience since	1965
Valves in operation	506
Valve hours	1.85×10^7 hrs
Failures	1

The stop and control valve designs have been changed and improved during the past fifteen years; therefore, arriving at a realistic historical failure rate applicable to our present designs requires a correct selection of experience. To select a representative sample of valves with previous, but similar design, we made the following assumptions:

a. For HP stop and control valves, the change from stem packings to sealing bushings is the starting point of our historical data. All HP top and control valves with sealing hushings built by Siemens and KWU are included in the data.

b. For IP stop and control valves of fossil reheat turbines with the relative low steam pressure, the sealing design is not as important a design criteria. Therefore, another criteria had to be chosen. As shown in Drawing NC 4.08, basically two different IP stop and control valves have been used in the past. The valve with a combined seat area for one of the two control valve seats and for the stop valve seat was the older design "B" and was replaced by design "A" with separate seat areas for the stop and control valves. We started our data with the introduction of this newer IP stop and control valve design "A" in 1965. All IP stop and control valves for fossil reheat turbines of design "A" and the present design built by Siemens and KWU are in the data.

All present HP and IP stop and control valves for fossil and nuclear applications are of the same basic design and fabricated under the same quality control program. IP valves for fossil applications and HP valves for LWR applications operate under similar conditions; whereas, the HP valves of fossil turbines operate under more severe conditions, therefore, it is a conservative approach to include all valves of present design in our statistical data to calculate the valve failure rates.

The Drawings TC 2.10b and TC 2.11b show our present HP and IP valve designs for fossil applications. The Drawing NC 2.05c shows our present HP stop and control valve design for LWR applications which is described in the speed control report No. E415. Some of the improvements of the present design are:

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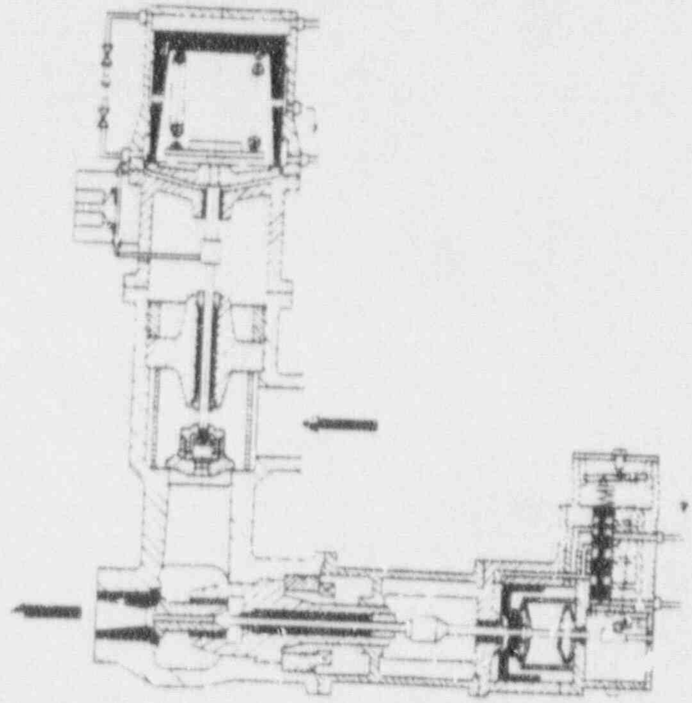
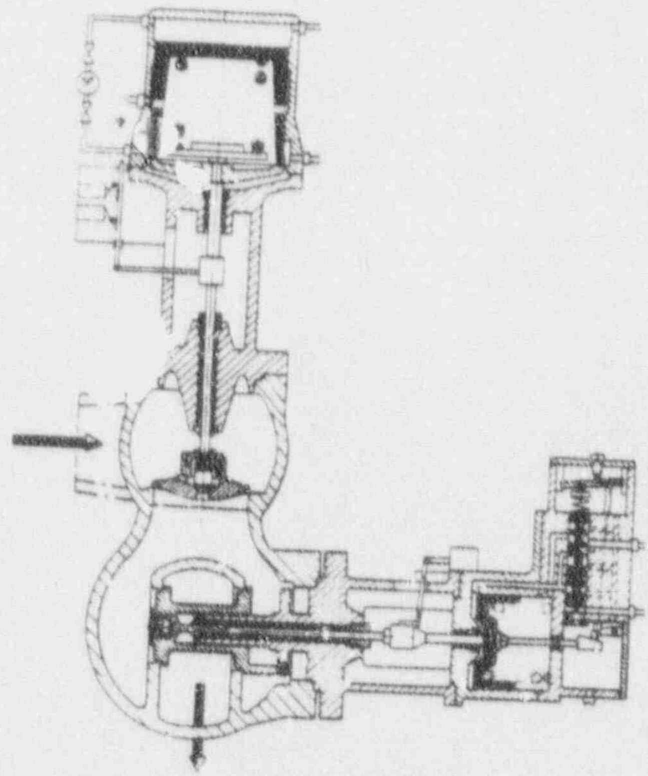
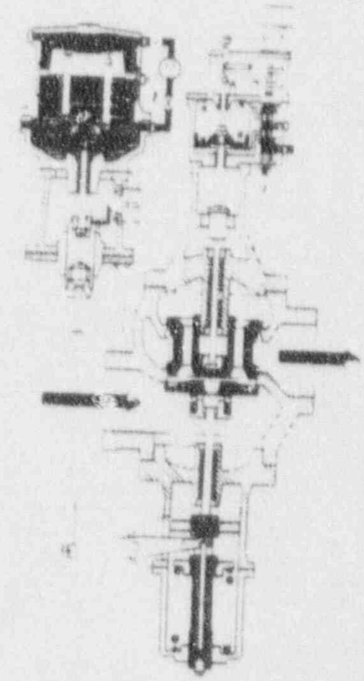


PREVIOUS HP STOP AND CONTROL VALVE DESIGN
WITH THE STEAM STRAINER IN THE STOP VALVE CASING



PREVIOUS IP STOP AND CONTROL VALVE
DESIGN A WITH DOUBLE SEAT CONTROL VALVE



PREVIOUS IP STOP AND CONTROL
VALVE DESIGN B WITH COMBINED
SEAT AREA FOR THE STOP AND
CONTROL VALVE

PREVIOUS STOP AND CONTROL VALVE DESIGN	ALLIS-CHALMERS POWER SYSTEMS INC
	NC 4.08

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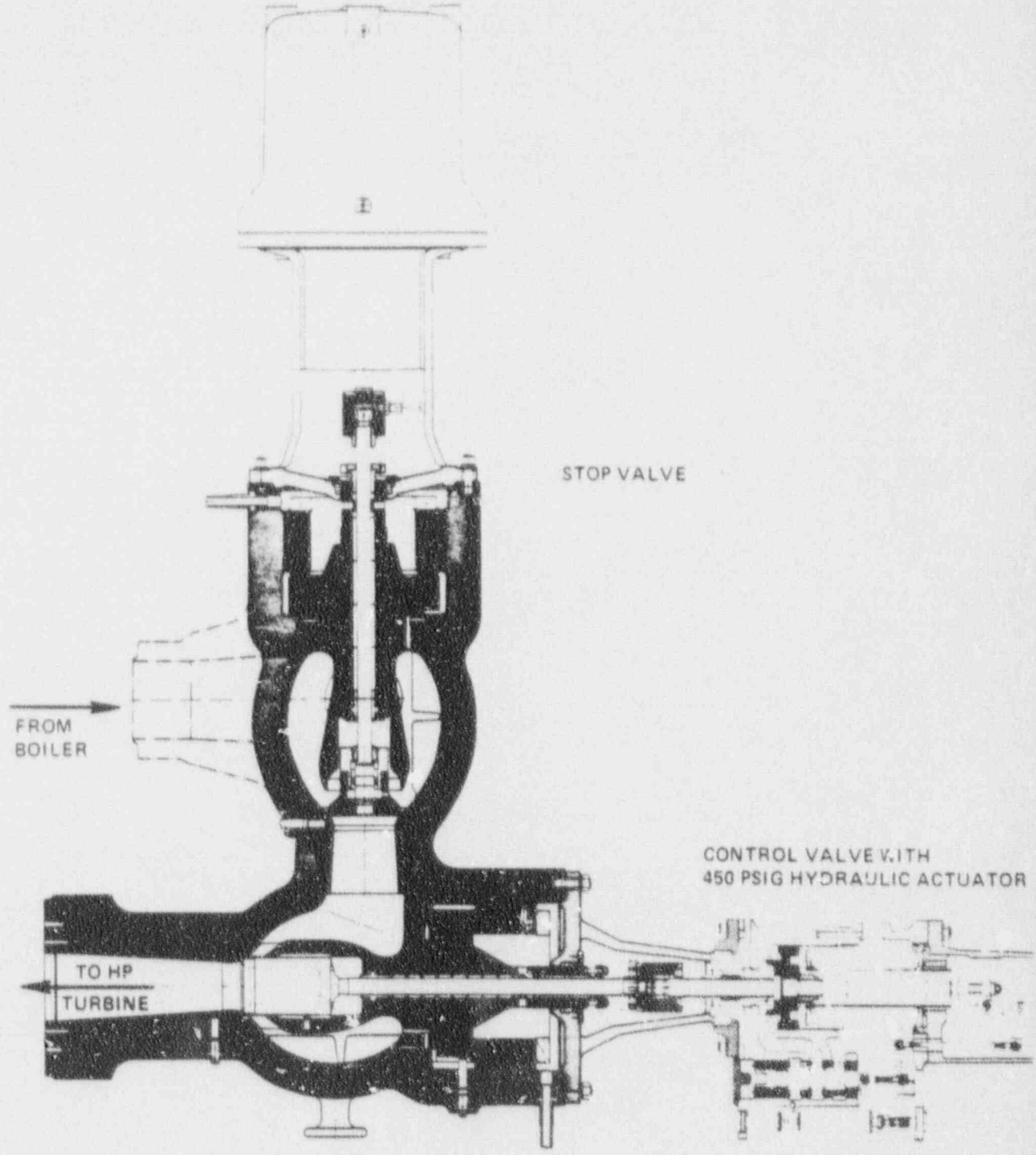

HP STOP AND
CONTROL VALVES FOR
FOSSIL AND HTGR APPLICATIONS

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TC 2.10b

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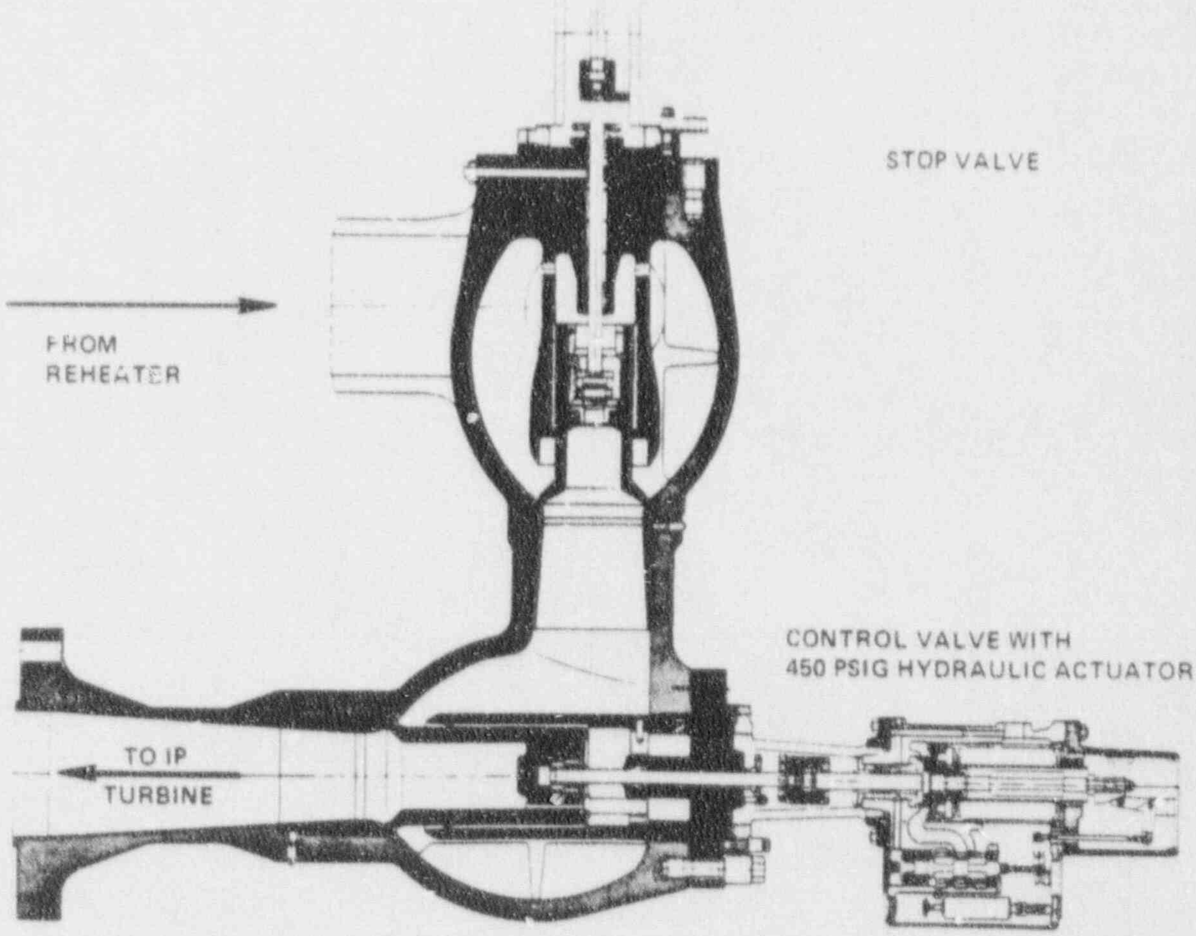

IP STOP AND
CONTROL VALVES FOR
FOSSIL AND HTGR APPLICATIONS

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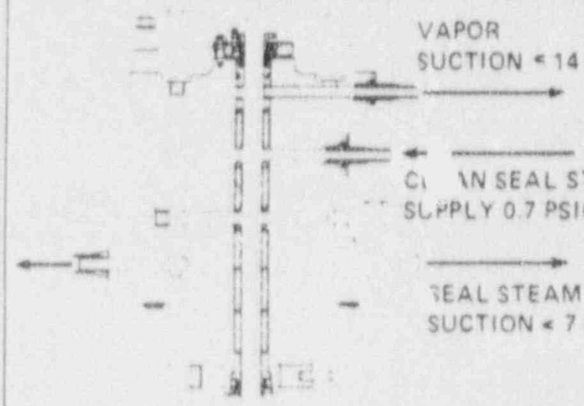
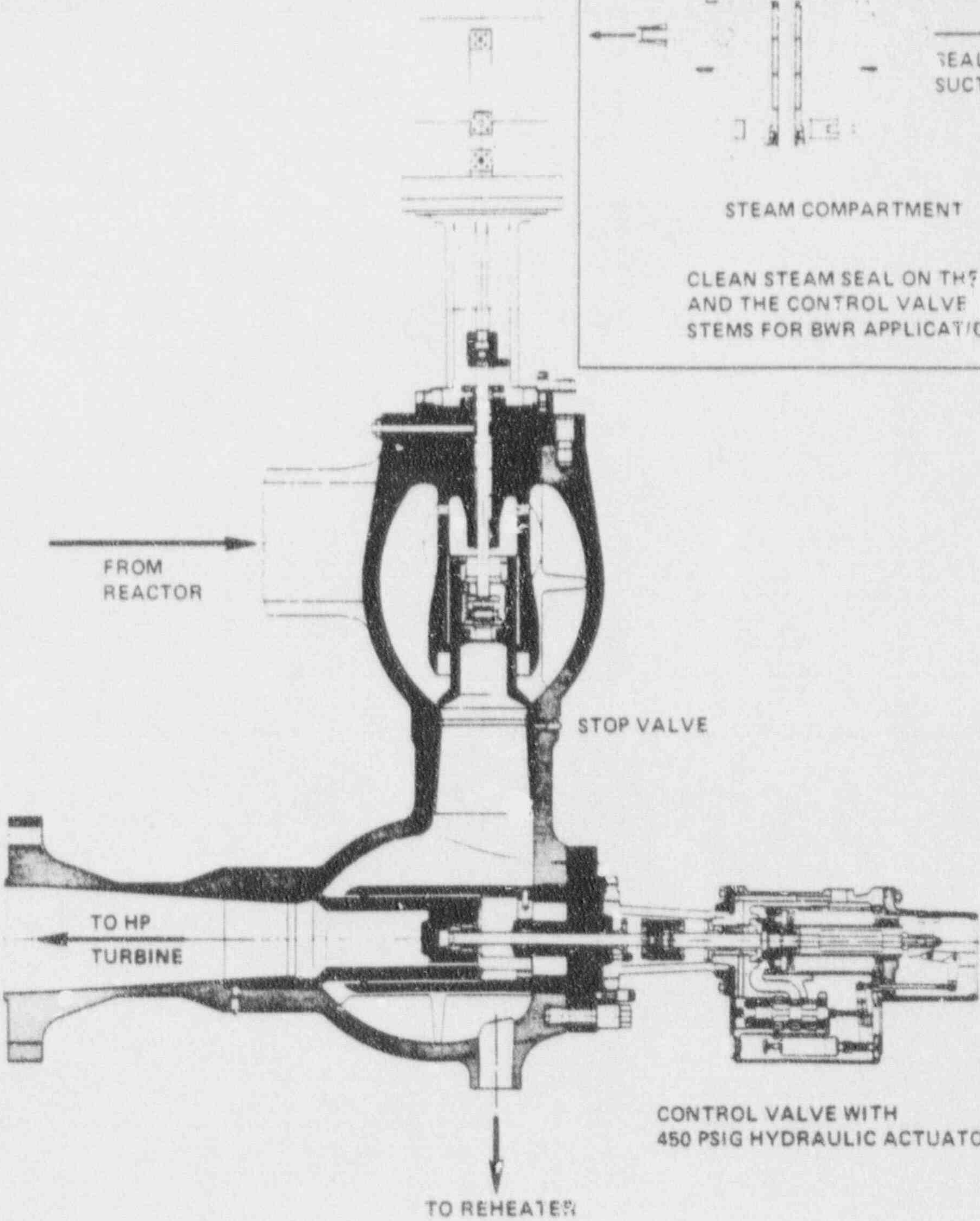


CLEAN STEAM SEAL ON THE STOP
AND THE CONTROL VALVE
STEMS FOR BWR APPLICATIONS



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HP STOP AND CONTROL
VALVE FOR BWR
AND PWR APPLICATIONS

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NC 2.05c

- Separate steam strainers
- Stop and all control valve cones guided (except HP stop valves for ≥1800 psig main steam)
- HP and IP stop valves with cones covered in open position
- Single seat control valves
- Control valves with two-step amplifiers with 450 psig fluid pressure actuator

Two (2) overspeed-type failures are known for the above specified sample of HP and IP stop valves, and one (1) failure for the sample of HP and IP control valves. All three failures happened on the same fossil fueled turbine-generator unit in 1967. The first stop valve failure happened because riveted support rods in the steam strainer broke, and parts of these rods prevented closing of the stop valve (see strainer Drawing NC 4.09 and previous HP stop and control valve Drawing NC 4.08, page 49). The steam strainer was replaced, however, the same type of failure happened again six weeks later. This time steam strainer parts also got into the control valve and blocked closure of the control valve too. These two stop valve failures and one control valve failure were due to failures of the steam strainer and not failures of the valves themselves. The steam strainer was designed with riveted support rods and local stress concentrations led to a fracture of the rod ends. Since then the strainer design has been completely changed, and in the current design heavy support bars are used to form a rigid welded frame, instead of the riveted rods. Furthermore, the steam strainers are no longer located within the stop valve casings, but rather in their own separate strainer casings (see Drawing NC 4.09) installed upstream of each stop valve in the main and reheat steam pipes.

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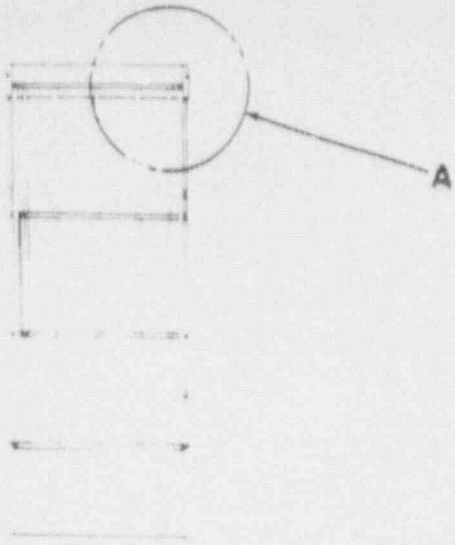
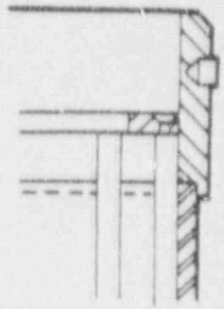
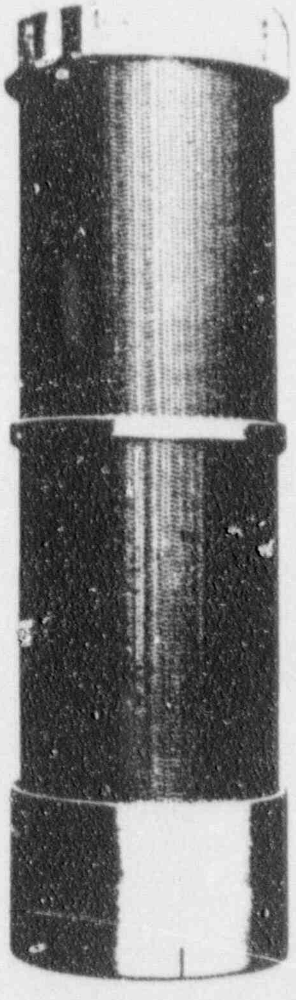
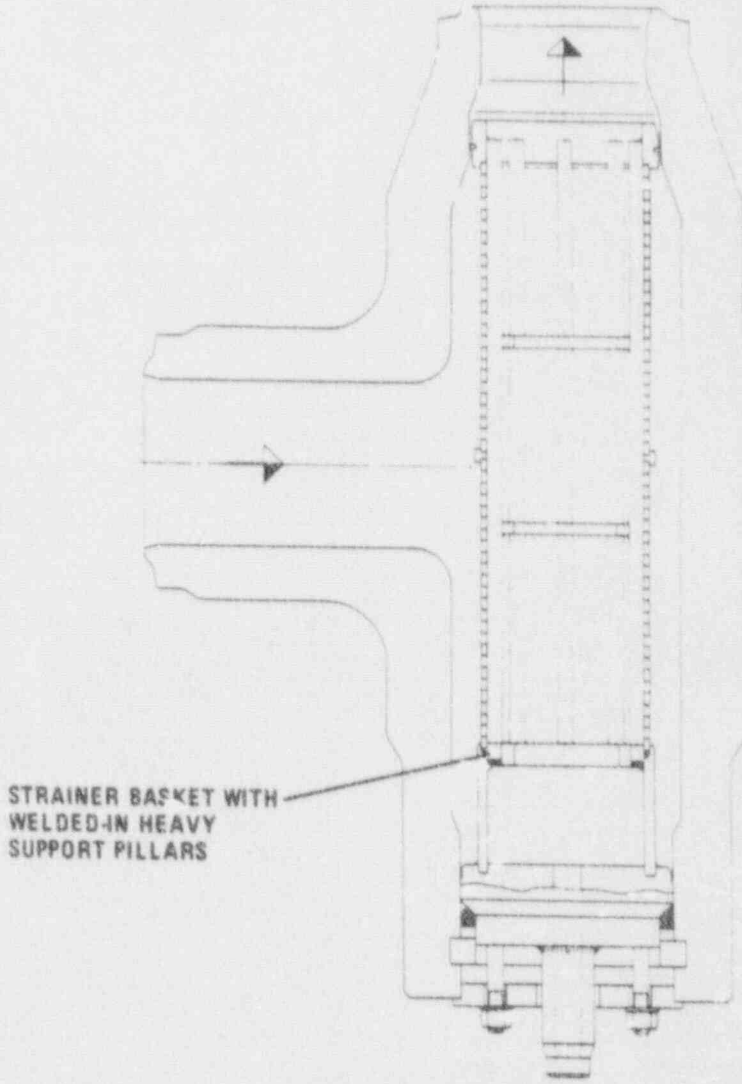




DETAIL A

PREVIOUS STEAM STRAINER DESIGN LOCATED IN THE STOP VALVE CASING WITH RIVETED SUPPORT RODS





SEPARATE STEAM STRAINER LOCATED IN THE
MAIN AND REHEAT UP STREAM OF THE STOP VALVES

STEAM STRAINER DESIGN	ALLIS-CHALMERS POWER SYSTEMS, INC.
	NC 4.09

These three failures are the only ones known for the selected sample of HP and IP stop and control valves which could lead to a >120% speed incident. There are no reported failures due to bent or sticking valve stems preventing valve closure.

Based on the foregoing, the failure rates and times for the stop and control valves are as follows:

<u>Stop Valves</u>

Stop valve hours	5.3×10^7 hrs
Historical failures	2
Assumed failures with 95% confidence	6.25
t, mission time	336 hrs (bi-weekly test)
λ, failure rate	$0.117/10^6$ hrs

<u>Control Valves</u>

Control valve hours	1.85×10^7 hrs
Historical failure	1
Assumed failures with 95% confidence	4.70
t, mission time	336 hrs (bi-weekly test)
λ, failure rate	$0.254/10^6$ hrs

Also included in the historical data of turbine valves are the butter fly-type LP valves of nuclear units for LWR application.

The butterfly-type LP stop valves and LP control valves are built with the same design criteria, as well as manufacturing and quality control procedures, as the HP stop and control valves.

The Drawing E5.127a illustrates the steam compartment and mechanical drive arrangement which is exactly the same for the LP stop valves and LP control

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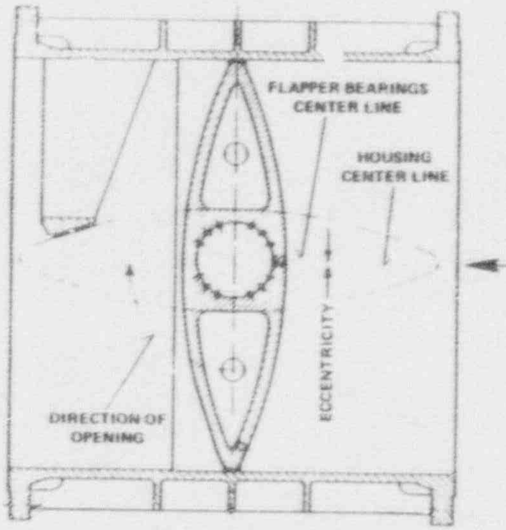
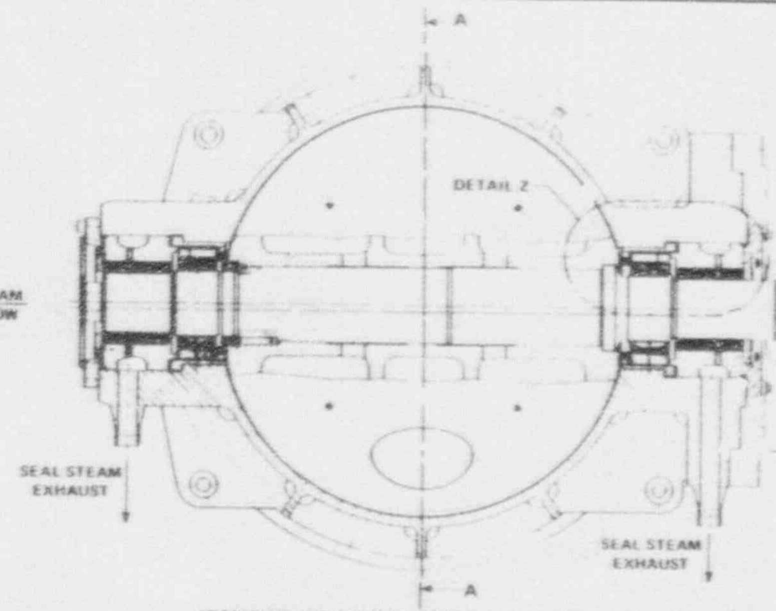
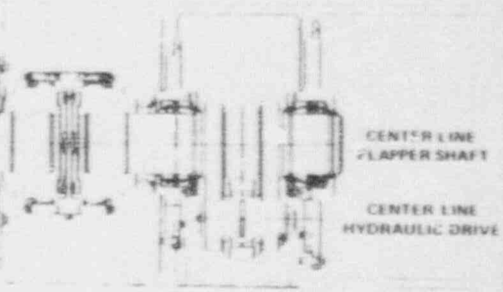


SECTION A-A

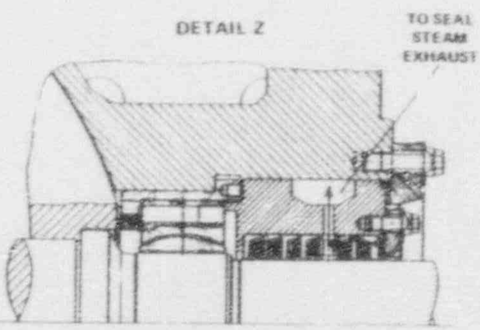




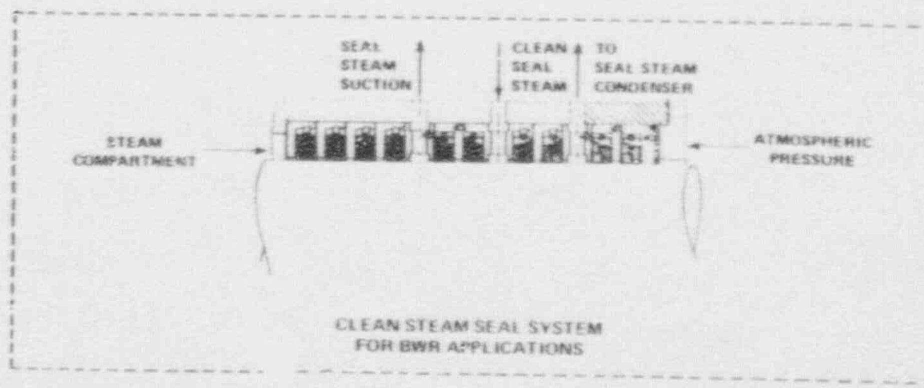
COUPLING AND MECH. DRIVE

IDENTICAL BUTTERFLY-TYPE VALVE DESIGN FOR LP STOP VALVES AND LP CONTROL VALVES

DETAIL Z



BEARING AND SHAFT SEAL PWR APPLICATIONS



CLEAN STEAM SEAL SYSTEM FOR BWR APPLICATIONS

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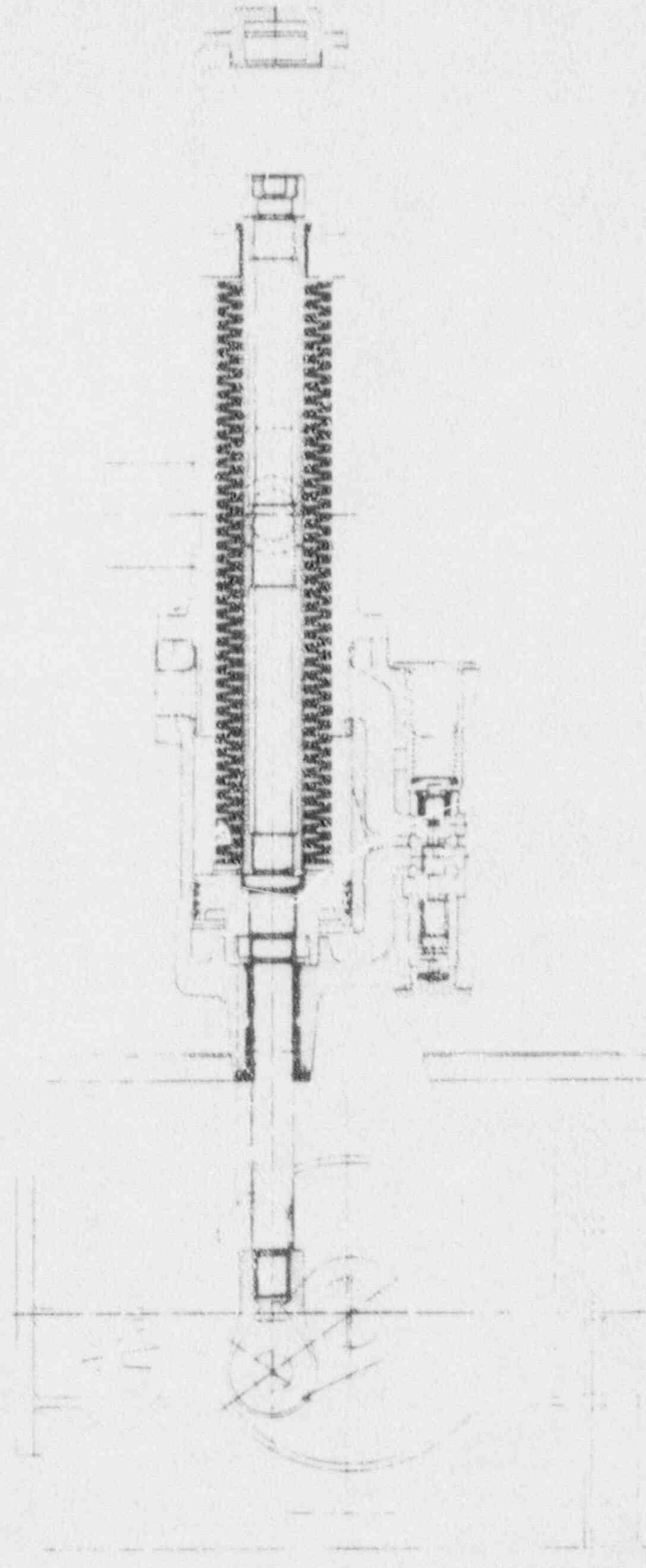
BUTTERFLY TYPE VALVE DESIGN FOR BOTH LP STOP VALVES AND

valves. The butterfly flaps have a circumferential clearance of about 1/20 inch to avoid valve sticking. The maximum possible steam leakage through the valve flap clearance is taken into account for safe turbine-generator full-load rejections to auxiliary load and no-load condition without tripping the turbine. The butterfly-type valves have a symmetrical design with a symmetrical bearing arrangement to eliminate axial thrust forces. The axial clearances and thrust bearing arrangement allow operation under all transient conditions without increased axial friction or excessive axial forces. Spherical journal bearings guarantee a minimum of rotational friction. Each flap journal bearing is designed with two separated bearing sliding surfaces, both equipped with low-friction material; this allows proper operation even in case of sticking of one sliding surface in each bearing. The valve spring and steam forces are acting in closing direction. The valve flap bearings are located eccentrically to the valve housing, resulting in proper steam forces in the closing direction. During load operation of the turbine-generator, the LP stop and control valves are fully open.

In the open-valve position, the hydraulic drive pulls the valve flap against an opening position support, thus avoiding valve flap vibrations. The mechanical valve drive like the valve flap itself is supported by two separate bearings. A full-flex gear-type coupling is provided to connect the flap shaft with the mechanical drive.

The hydraulic actuators are shown in Drawing NC 4.21 for the LP stop valves and in NC 4.22 for the LP control valves. The hydraulic drive of the LP control valve is of the same design as the HP and IP control valve actuators with two-step amplifiers. On the LP stop valves, the two-step amplifiers are replaced



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ACTUATOR OF A
BUTTERFLY - TYPE
LP STOP VALVE

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NC 4.21

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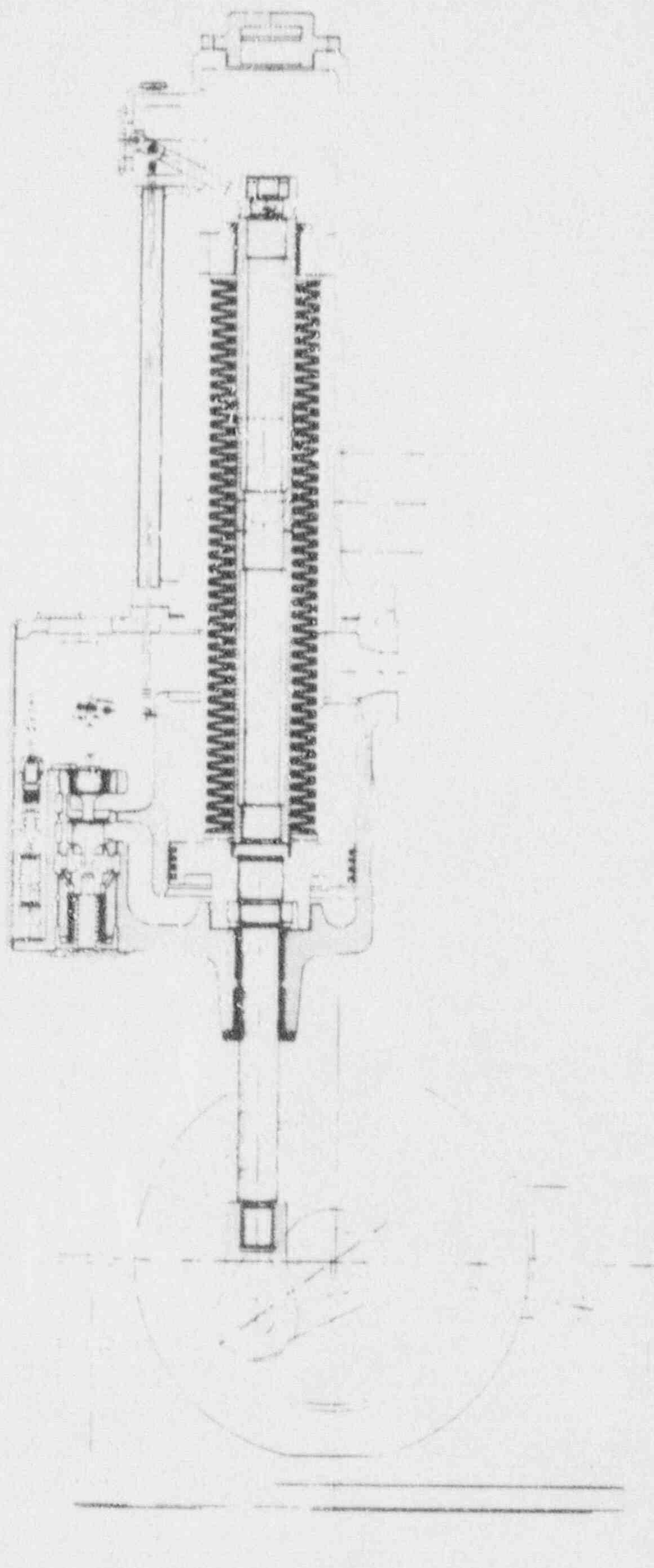


ACTUATOR OF A
BUTTERFLY - TYPE
LP CONTROL VALVE

ALLIS-CHALMERS
POWER SYSTEMS, INC.

NC 4.22

by a fast response control piston. This feature provides the LP stop valves with a highly reliable two-position (fully open - fully closed) function.

Similar to the HP stop and control valves, the butterfly-type LP stop and control valves are provided with automatic test features for the bi-weekly valve test.

4.3.2 Follow-Up Pistons, Elements E 1-6, F 1-6 and H 1 & H 2

The applicable historical data is:

Operation experience since	1954
Pistons in operation	1360
Piston hours	7.2×10^6 hrs
Failures	0

The follow-up pistons are simple adjustable drain pistons. During the past 20 years there has been no failure of such a follow-up piston in closing direction which would result in a faulty opening signal to the control valves. The fail safe direction of the follow-up pistons is the drain opening or control valve closing direction. The failure rate and mission time for these pistons are as follows:

Follow-up piston hours	7.2×10^6 hrs
Historical failures	0
Assumed failures with 95% confidence	2.95
t, mission time	8760 hr (yearly inspection)
λ, failure rate	$0.041/10^6$ hrs

4.3.3 Electro-Hydraulic Converters without Coils, Elements G 1 & G 2

The applicable historical data is:

Operation experience since	1963
EH converters in operation	86
EH converter hours	2.65×10^6 hrs
Failures	0

Since introduction of the EHC in 1963, no failures in control valve opening direction are reported for the mechanical-hydraulic part of the EH converter. The electrical part (coil) is included in the steam admission control. The converter is not a new design because the hydraulic part was used as a hydraulic amplifier for turbines with only Mechanical-Hydraulic Control (MHC), and with the present EHC system, the EH converter receives both the electrical input signal from the admission control and the hydraulic signal from the MHC. The failure rate and mission time for the EH converter without coil are:

EH converter hours	2.65×10^6 hrs
Historical failures	0
Assumed failures with 95% confidence	2.95
t, mission time	8760 hr (yearly inspection)
λ, failure rate	$1.11/10^6$ hrs

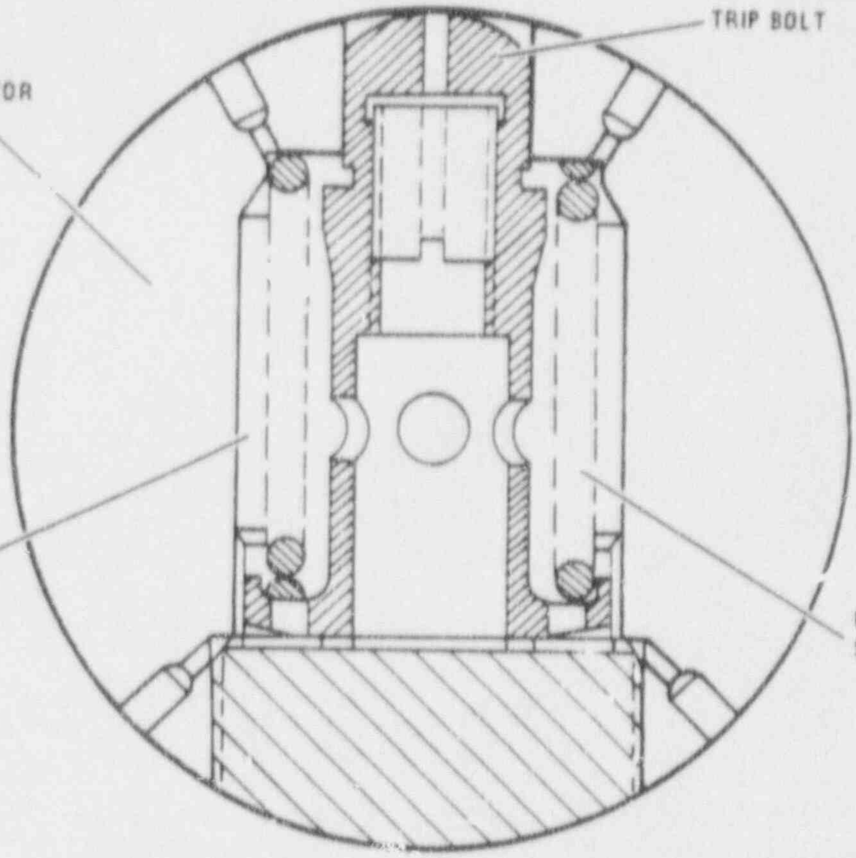
4.3.4 Mechanical-Hydraulic Control (MHC), Element 1

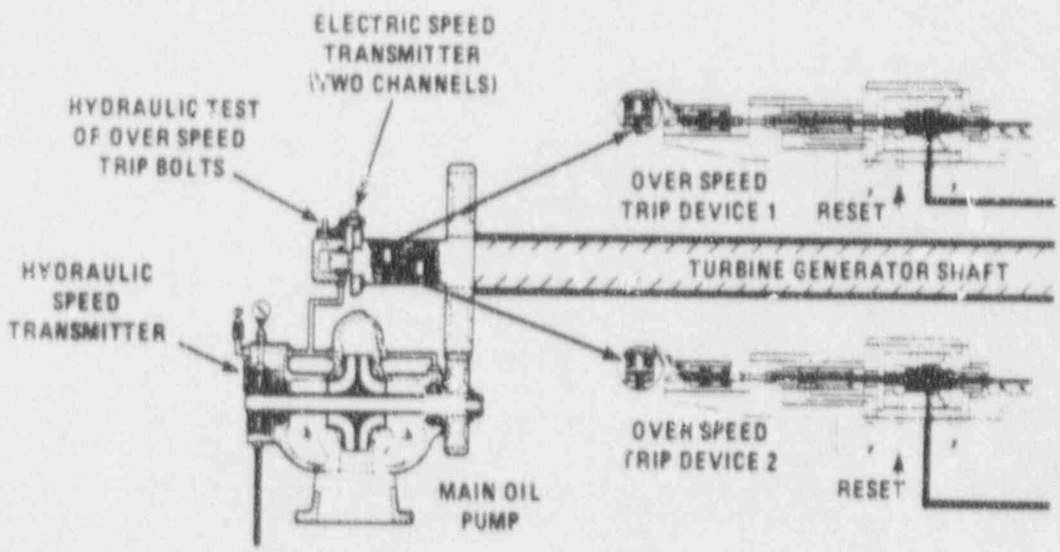
The applicable historical data is:

Operation experience since	1954
MHC's in operation	427
MHC hours	1.68×10^7 hrs
Failures	1

One failure of the above MHC's in the control valve opening direction has been reported. The hydraulic speed transmitter did not receive a sufficient oil supply from the turbine main shaft oil pump (see Drawing E5.121). The low oil flow supply resulted in a too low discharge pressure of the impeller. If, during this condition, a real overspeed event would have occurred, the increase in oil pressure from the speed transmitter would not have been high enough to prevent an overspeed. The failure rate and mission time are:

MHC hours	1.68×10^7 hrs
Historical failures	1
Assumed failures with 95% confidence	4.7
t, mission time	8760 hr (yearly inspection)
λ, failure rate	$0.28/10^6$ hrs





OVERSPEED TRIP BOLTS	ALLIS-CHALMERS POWER SYSTEMS INC
	E5.121

4.3.5 Admission Controls with Coils, Elements K 1 & K 2

The applicable historical data is:

Operation experience since	1963
Admission controls in operation	86
Admission control hours	2.65 x 10^6 hrs
Failures	4

Three of the four failures were open-circuit failures of the plunger coil in the EH converter, which happened during turning gear operating caused by extremely high amplitude oscillations of the electrical EHC signal. This problem was eliminated for future units by filtering the electrical low speed signal. With the new design of this component, failures of three spring type connections (which normally provide one-out-of three reliability) to the plunger coil must fail to produce this event.

The fourth admission control failure was a broken electrical lead in the the EHC cabinet. All four failures occurred in control valve opening direction and could have led to an overspeed event. The failure rate and mission time are:

Admission control hours	2.65 x 10^6 hrs
Historical failures	4
Assumed failures with 95% confidence	9.1
t, mission time	8760 hr (yearly inspection)
λ, failure rate	3.43/10^6 hrs

4.3.6 EHC speed Control and Speed Measuring Device, Elements L & M & N

The applicable historical data is:

Operation experience since	1963
Speed controls in operation	86
Speed control hours	2.65×10^6 hrs
Failures	0

These elements consist of the EHC speed control with digital/analog converter, the speed measuring device including the disc with 120 permanent magnets mounted on the HP turbine rotor and the two channels of the EHC speed transmitter, pulse converter and time supervisory subsystem. There have been no reported failures in the control valve opening direction for all of these elements. The failure rate and mission time are:

Speed control hours	2.65×10^6 hrs
Historical failures	0
Assumed failures with 95% confidence	2.95
t, mission time	8760 hrs (yearly inspection)
λ, failure rate	$1.11/10^6$ hrs

4.3.7 Power Supply, Elements O & P

The applicable historical data is:

Operation experience since	1963
Power supplies in operation	86
Power supply hours	2.65×10^6 hrs
Failures	0

For a high reliability of the EHC, we require one primary ac power source, and either one or two independent dc power sources depending on the reliability and characteristics of the primary source. There are separation diodes and an ac/dc as well as a dc/dc converter in the EHC cabinet for the internal power supply. The failure rate of the power sources cannot be determined by A-CPSI since we do not generally design or furnish the power sources. However, in our total experience with EHC, we know of no failure in the internal supply, and we also know there has never been a case of loss of all power sources during operation of a turbine generator with EHC. Despite the limited operating experience we feel it is sufficiently conservative to use our historical data of zero failures for the power supply and power source and to calculate the failure rate with 95% confidence level:

Power supply hours	2.65×10^6 hrs
Historical failures	0
Assumed failure with 95% confidence	2.95
t, mission time	8760 hrs (yearly inspection)
λ, failure rate	$1.11/10^6$ hrs

4.3.8 Change-Over Device, Element R

The applicable historical data is:

Operation experience since	1966
Change-over devices in operation	53
Change-over device hours	1.45×10^6 hrs
Failures	0

During normal turbine-generator operation this device is in its safe position, and also does not have to act in case of a turbine trip. It is only used during periodic testing of the trip systems. This means the probability of an overspeed failure due to the change-over device is extremely small, and the historical data yields an unrealistically high failure rate for this device because of the relatively small experience time.

The change-over device, shown in Drawing NC 4.07, is mainly the change-over valve which conducts the trip fluid from the main trip valves to the turbine stop and control valves. During normal operation the solenoid valves, A, B and C are in their safe (not energized) position and the lower chamber of the change-over valve received control fluid from solenoid valve A, while solenoid valves B and C are draining the chamber of the change-over valve.

During the bi-weekly turbine trip test with the Automatic Turbine Tester (ATT), the solenoid valves B and C are positioned to supply control fluid to the upper chamber and the solenoid valve A drains the lower chamber. The change-over valves moves downward and the trip signal from the normal overspeed trip devices to the stop and control valves is interrupted. However, before this change-over is initiated, electrical trip circuits are connected to the solenoid valves B and C to trip the turbine in the event that a real overspeed should occur during testing. Part of the ATT program is a test of these electrical overspeed trip circuits and a start of the trip device test program is only possible after a successful test of these electrical overspeed trip circuits. The electrical overspeed trip

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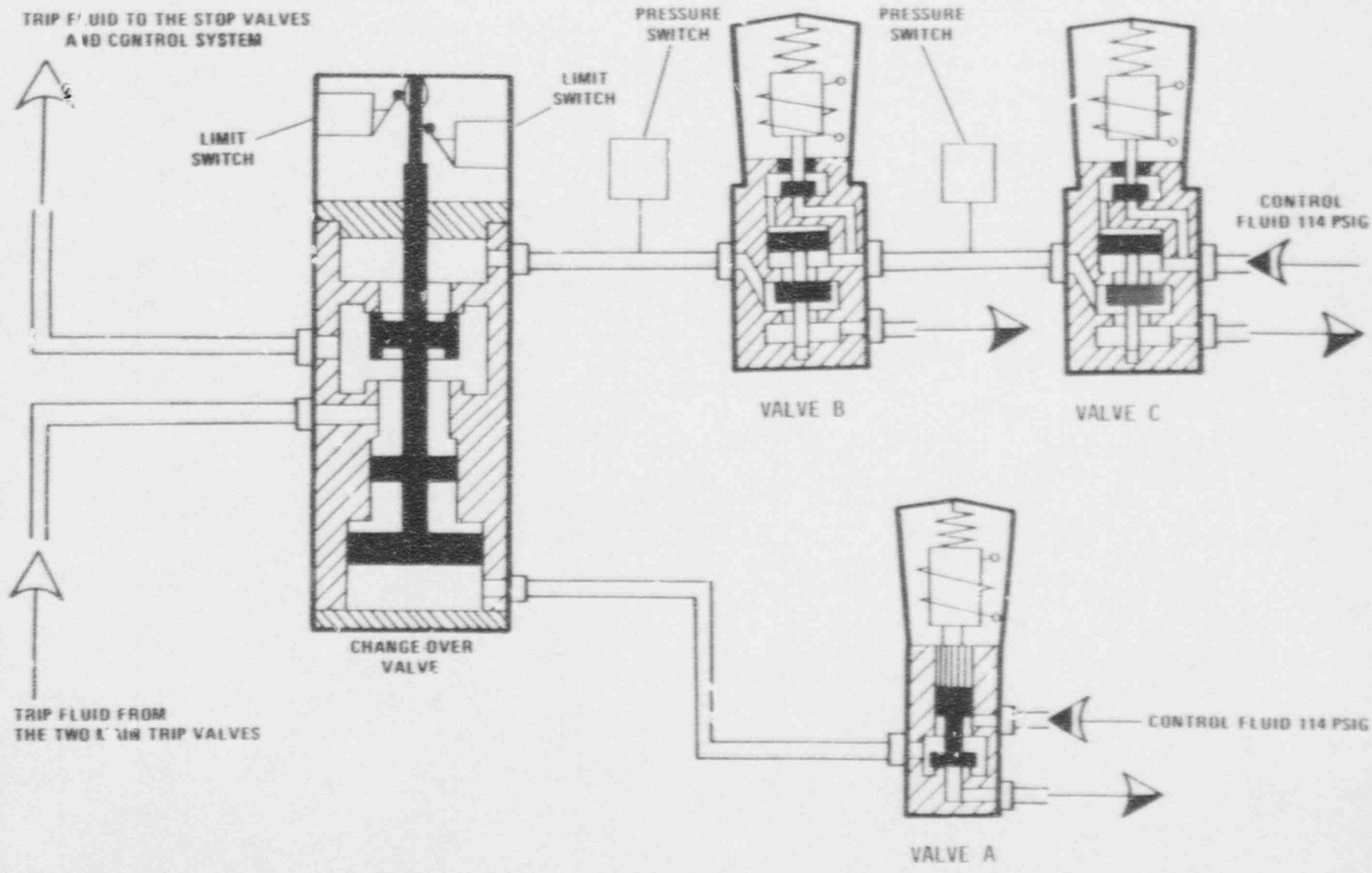

ALL VALVES ARE SHOWN IN TURBINE OPERATING CONDITION.
TO AVOID A REAL TRIP THE CHANGE-OVER VALVE HAS TO
BE MOVED INTO THE CLOSING DIRECTION (DOWNWARDS)

CHANGE-OVER DEVICE
FOR TESTING THE TURBINE
TRIP DE 'ICES

ALLIS-CHALMERS
POWER SYSTEMS, INC.

NC 4.07

is a one-out-of-two signal to the one-out-of-two solenoid valves B and C. After testing, the change-over valve is reset in its normal position by opening the control fluid supply with solenoid valve A and draining the upper chamber with solenoid valves B and C. Very high reliability for the resetting of the change-over valve is provided by checking the position of the change-over valve with a limit switch, and by two pressure switches sensing the control fluid pressure decrease behind the solenoid valves B and C.

For practical purposes, the failure mode of this change-over device is a simultaneous failure of all three solenoid valves. The average failure rate of solenoid valves is approximately 10 per 10^6 hrs per Reference (11), table D.1. This failure rate is very conservative for our case because the solenoid valves need not act during a real turbine trip; they only have to stay in the safe (not energized) position.

Since all three solenoid valves must fail, the failure rate of the change-over device is calculated as a redundant system consisting of three solenoid valves as follows:

t, mission time	336 hrs (bi-weekly test)
λ, failure rate of a solenoid valve	$10/10^6$ hrs
EFR, the effective failure rate of the one-out-of-three solenoid valve system used as a model for the change-over device for mission time of 336 hours	$0.0001123/10^6$ hrs

This is calculated as follows:

$P_{system} = (P_{sol.\ valve})^3$ = failure probability of 1-out-of-3-valves

$EFR \cdot t = (\lambda_{sol.\ valve} \cdot t)^3$

$EFR = (\lambda_{sol.\ valve})^3 \cdot t^2$

$EFR = (10/10^6\ hr)^3\ (336\ hr)^2$

$EFR = .0001129/10^6$ hr (effective failure rate for t = 336 hrs)

$EFR = .0001123/10^6$ hr (exact computed value for the 336 hrs mission time)

4.3.9 Main Trip Valve, Elements S 1 and S 2

The applicable historical data is:

Operation experience since	1958
Main trip valves in operation	278
Main trip valve hours	1.43×10^7 hrs
Failures	0

In case of a trip the main trip valves drain the trip fluid pressure initiating a closing of the stop and control valves. The main trip valves are arranged in a one-out-of-two system and are spring loaded in draining direction. An auxiliary trip fluid pressure holds the main trip valves in closing position. This pressure will be released if one of the trip signals actuates a trip device and the main trip valves move by spring force to their draining position. Considering this simple reliable device, it is very conservative to use the historical data with a confidence level of 95%:

Main trip valves in operation	278
Main trip valve hours	1.43×10^7 hrs
Historical failures	0
Assumed failures with 95% confidence	2.95
t, mission time	336 hrs (bi-weekly test)
λ, failure rate	$0.206/10^6$ hrs

4.3.10 Overspeed Trip Test and Resetting Device, Element T

The applicable historical data is:

Operation experience since	1958
Overspeed trip test devices in operation	242
Overspeed trip test device hours	1.4×10^7 hrs
Failures	0

Like the Change-Over Device Element R, the "Overspeed Trip Test and Resetting Device" T is only used for testing, and for resetting the trip devices after shutdown and testing (see Drawing E5.122a). During normal turbine-generator operation the overspeed trip test and resetting device is in its safe position and need not act in case of a real trip.

The manual overspeed trip test device and the manual overspeed trip reset till be used only for local testing at the turbine. Operating instructions call for a yearly test or test after an inspection of the

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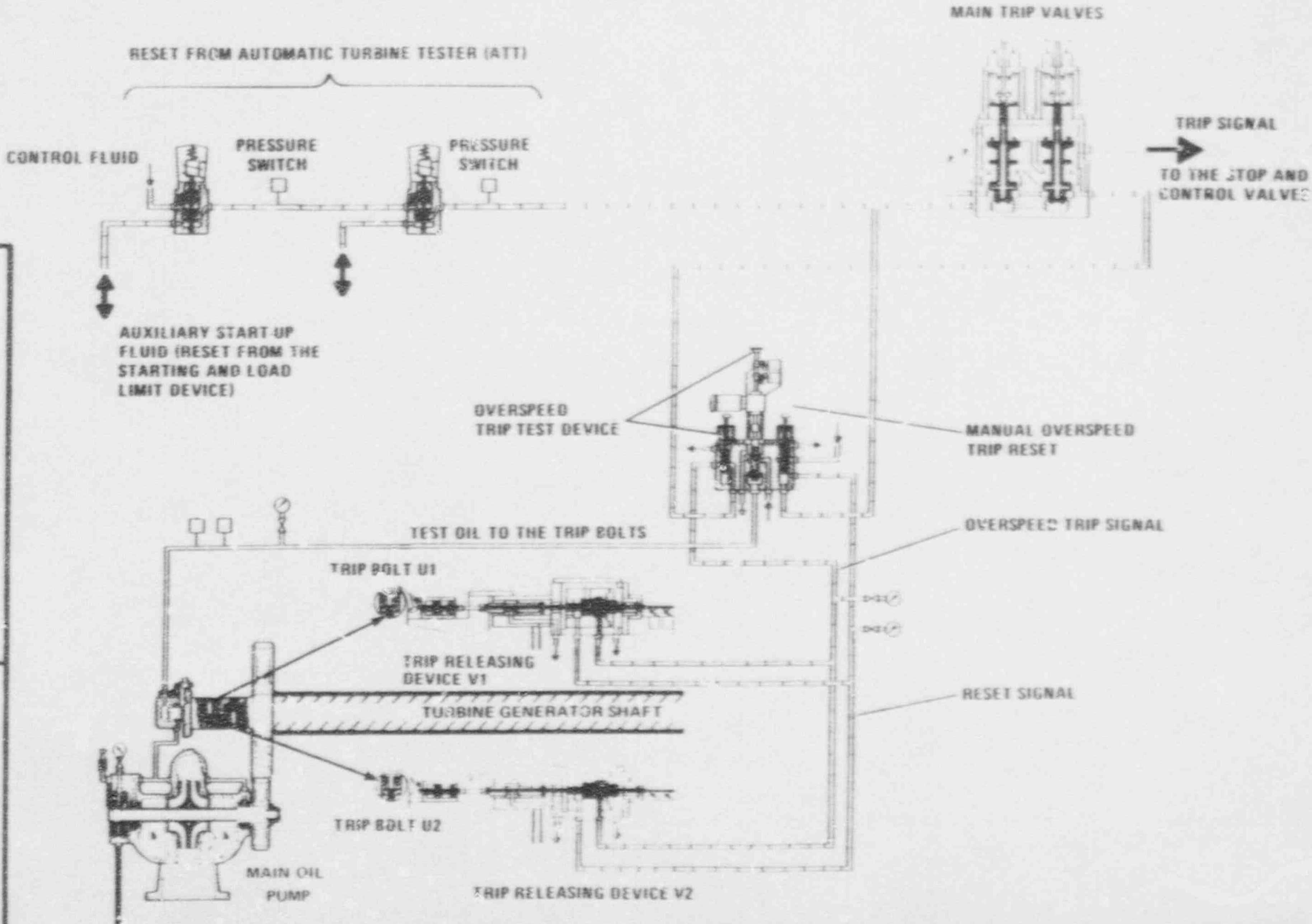


OVERSPEED TRIP TEST
AND RESETTING DEVICE

ALLIS-CHALMERS
POWER SYSTEMS INC

E5.122a

trip system only. The test is required to check the adjustment of the two pressure switches of the test oil supply to the trip bolts. The setting of these pressure switches during automatic testing with the ATT indicate at rated speed the proper function of the two trip bolts.

A malfunction of this manual trip test and resetting device in the stop and control valve opening direction is very unlikely because the pistons are spring-loaded to return to their safe position after they were pushed down for testing. Testing is only yearly and locally at the turbine. The two pistons are supplied with key-locking devices with which the pistons have to be locked in their safe position except when required for test.

For this extremely safe manual trip test and resetting device, we assume the same effective failure rate level as for the Change-Over Device R because all the above precautions against malfunctions make the manual overspeed trip test and resetting device as safe as the change-over device. Since there are two pistons, the effective failure rate of the manual overspeed trip test and resetting device is two times the rate of the change-over device:

t, mission time	8760 hrs (yearly test)
EFR, effective failure rate of the manual overspeed trip test and resetting device	$0.000225/10^6$ hrs

The second function of the overspeed trip test and resetting device is the remote resetting of the trip devices including the main trip valves and overspeed trip release devices. These functions will be used bi-weekly during each ATT test. The fluid supply for the resetting comes either from the starting and load limit device as auxiliary start-up fluid, or as control fluid. Due to the design of the starting and load limit device, the auxiliary start-up fluid line can only be pressurized when no pressure to the control valves is available. The control fluid is available whenever the control fluid pumps are running; therefore, it is reasonable to consider only the control fluid supply because the failure probability due to an accidental control fluid supply is many times larger than due to an accidental supply of auxiliary start-up fluid.

A malfunction of the two reset solenoid valves could prevent an overspeed trip; however, in their safe (not energized) position the control fluid cannot block a real trip. During reset the ATT opens the solenoid valves for the control fluid and closes them automatically. During the test and reset time the electrical overspeed trip described under "Element R" is active and would trip the turbine in a real overspeed event. The two reset solenoid valves will be closed in sequence by closing first the second supply valve (left valve in Drawing E5.122a, page 72), and checking the pressure decay between this solenoid valve and the main trip valves. This also indicates that no auxiliary start-up fluid pressure is built up. Then the first (right) valve will be closed and the pressure decay between the two valves will be checked. With this valve and pressure switch arrangement, it is nearly certain that the valves will be reset to their

safe (not energized) position, and cannot prevent a real overspeed trip. To take a conservative approach, we used a failure rate of $10/10^6$ hrs for each solenoid valve regardless of the fact that for a real trip the reset solenoid valves remain in their safe (not energized) position. The failure rate and mission time are as follows:

t, mission time	336 hrs (bi-weekly test)
Failure rate of a solenoid valve	$10/10^6$ hrs
λ, failure rate of the remote resetting device with one-out-of-two solenoid valves	$0.0335/10^6$ hrs

4.3.11 Overspeed Trip Bolts, Elements U 1 & U 2

The applicable historical data is:

Operation experience since	1958
Trip bolts in operation	486
Trip bolt hours	2.78×10^7 hrs
Failures	7

The operating experience includes all the trip bolts which are automatically or at least manually testable at rated speed. One of the seven failures was caused by ventilation suspended particles from an abnormally dirty environment which entered the front bearing housing and prevented proper operation of the trip bolts. The remaining six failures were all due to fretting corrosion, which resulted in a sticking of the trip bolts.

The trip bolt design is shown in Drawing E5.121, page 63. Under certain conditions such as large shaft vibrations and aggressive air or oil, there can be a tendency to form fretting corrosion. The best method to detect sticking of the trip bolts due to fretting corrosion or dirt is the regular test of the trip bolts which includes with the ATT an automatic check that the oil pressure level at which the trip bolts move into the trip position is correct. All seven trip bolts failures were found during manual testing, and there are no reported failures of this type on a turbine equipped with ATT. Although we have no proof, we believe that the failed trip bolts were not tested as frequently as every two weeks, as is recommended for units equipped with the ATT. During the past 15 years our large turbine-generators were equipped with two separate trip bolts and separate trip release devices, and there is no case in which both overspeed trips failed at the same time or during the same test.

Regardless of the fact that bi-weekly testing should prevent future trip bolt failures which happened in the past, we will take the conservative approach that a failure in the future is as likely as failures in the past. Therefore, the failure rate for the trip bolts is:

Trip bolt hours	2.78×10^7 hrs
Historical failures	7
Assumed failures with 95% confidence level	13.1
t, mission time	336 hrs (bi-weekly test)
λ, failure rate	$0.471/10^6$ hrs

4.3.12 Trip Release Device, Elements V 1 & V 2

The applicable historical data is:

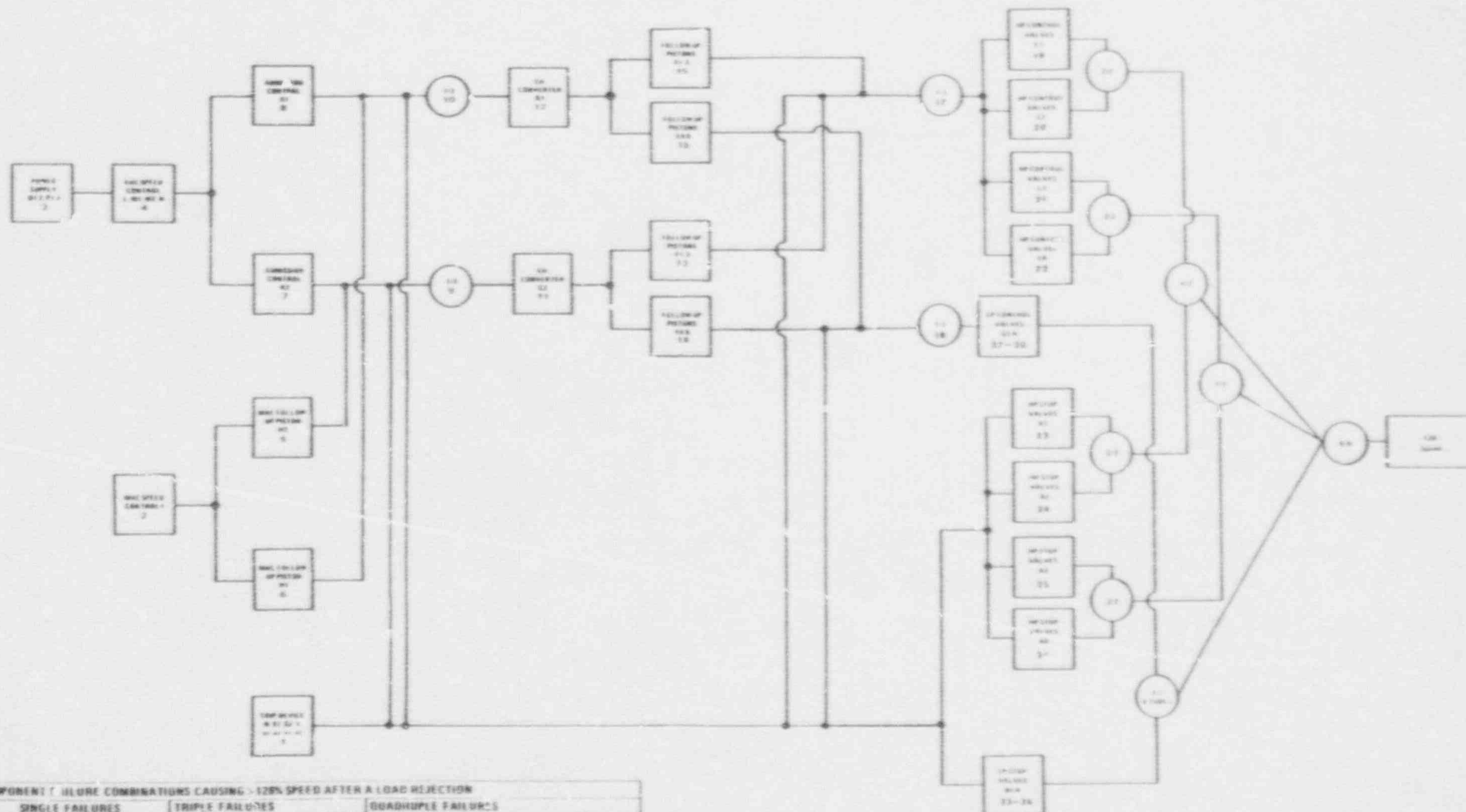
Operation experience since	1958
Trip release devices in operation	486
Trip release device hours	2.78×10^7 hrs
Failures	0

The trip release device is a simple hydraulic piston and spring system converting the movement of the trip bolt into a hydraulic signal. No failure of these devices has been reported, which results in the following failure rate:

Trip release device hours	2.78×10^7 hrs
Historical failures	0
Assumed failures with 95% confidence level	2.95
t, mission time	336 hrs (bi-weekly)
λ, failure rate	$0.106/10^6$ hrs

4.4 Overspeed Failure Probability Calculation

For purposes of computer calculation of the overspeed failure probability, the overspeed prevention system was arranged into system components 1 to 36 as shown in the reliability block diagram NC 5.02a which is equivalent to diagrams NC 5.01a, page 36 and E 5.120, page 41. It can be seen that some of the components 1 to 36 are formed from several elements (A to V). For example, the trip device component No. 1 consists of eight elements including the change-over device R, two main trip



COMPONENT [illegible] ILURE COMBINATIONS CAUSING >120% SPEED AFTER A LOAD REJECTION					
SINGLE FAILURES		TRIPLE FAILURES		QUADRUPLE FAILURES	
NONE		1 & 13 & 15	1 COMB	1 & 2 & 7 & (8 OR 12 OR 15 OR 16)	4 COMB
DOUBLE FAILURES		1 & 14 & 16	1 COMB	1 & 2 & 8 & (11 OR 13 OR 14)	3 COMB
1 & 19-22	4 COMB	1 & 2 & 3	1 COMB	1 & 3 & 5 & (6 OR 12 OR 15 OR 16)	4 COMB
1 & 27-30	4 COMB	1 & 2 & 4	1 COMB	1 & 3 & 6 & (11 OR 13 OR 14)	3 COMB
19-22 & 23-26	8 COMB	1 & 11 & 15	1 COMB	1 & 4 & 5 & (8 OR 12 OR 15 OR 16)	4 COMB
27-30 & 33-36	4 COMB	1 & 11 & 16	1 COMB	1 & 4 & 6 & (11 OR 13 OR 14)	3 COMB
DOUBLE FAILURE COMBINATIONS		1 & 12 & 13	1 COMB	1 & 5 & 7 & (12 OR 15 OR 16)	3 COMB
	28	1 & 12 & 1[illegible]	1 COMB	1 & 6 & 8 & (11 OR 13 OR 14)	3 COMB
		1 & 11 & 12	1 COMB	QUADRUPLE FAILURE COMBINATIONS	27
		TRIPLE FAILURE COMBINATIONS		QUINTUPLE FAILURE COMBINATION	1

SCHEMATIC RELIABILITY BLOCK DIAGRAM WITH FAILURE COMBINATIONS OF A 4 FLOW REHEAT TURBINE F[illegible] LIGHT WATER APP[illegible]

NC 5.07a

REVISION

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valves S 1 and S 2, the overspeed trip test and resetting device T, two overspeed trip bolts U 1 and U 2 and the trip releasing devices V 1 and V 2.

The computer program calculates the system failure probability on the basis of the following commonly known principles of reliability analysis:

1. Probability of failure:

   $P = 1-e^{-\lambda t}$ (for exponent $\lambda t > 0.3$)

   $P = \lambda t - \frac{(\lambda t)^2}{2!} + \frac{(\lambda t)^3}{3!} - \frac{(\lambda t)^4}{4!}$ (for exponent $\lambda t \leq 0.3$)

   where:

   λ = failure rate, failures per 10^6 hr

   t = mission time, hr

2. Series elements

   $P = P_1 + P_2 + \ldots$ (conservative equation because it assumes that events are mutually exclusive)

3. Parallel (redundant) elements

   $P = P_1 \times P_2 \times \ldots$

There is no single failure of any component in the system which could lead to a $\geq 120\%$ speed event. However, there are multiple failure possibilities which could lead to this event. The computer program evaluates all possible failure combinations

listed in Diagram NC 5.02a, page 78. The highest order of failure is one quintuple simultaneous failure which could lead to an overspeed event.

The list of multiple failures at the bottom of diagram NC 5.02a, page 78, shows clearly the importance of the components 1, and 19 to 36, which are the trip device, and the stop and control valves. Double failures of these components could lead to an overspeed event. The computer calculations show that the double failure combinations have the following influence on the overall failure probability results:

Double Failure Combinations	4-Flow LP Turbine Influence %	6-Flow LP Turbine Influence %
HP Stop Valves and HP Control Valves	13.22	17.65
HP Stop and Control Valves Cross-Connection	13.22	17.65
LP Stop Valves and LP Control Valves	13.22	26.48
Trip Device (1) and HP Control Valves	4.45	5.96
Trip Device (1) and LP Control Valves	4.45	8.96
Reverse Reheater Evaporation through HP Control Valves	28.71	-
Total Influence	77.27	76.70

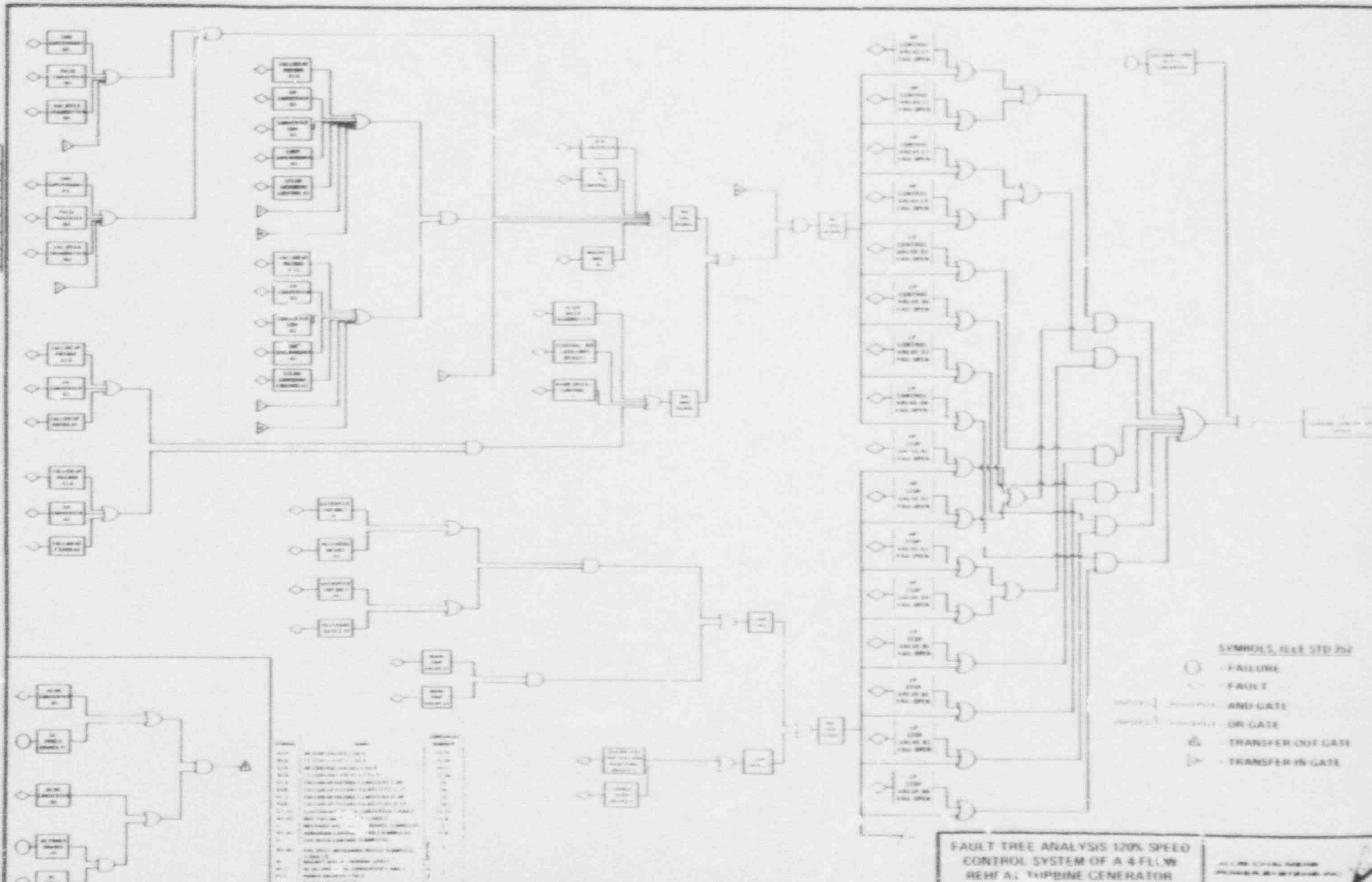
The remaining 22.73% of the 4-flow and 23.30% of the 6-flow LP turbine of the overall failure probability numbers are influenced by triple, quadruple and quintuple failure combinations including failures of the extraction system.

The failure probability of the overspeed prevention system was calculated for two different modes of operation; first, for turbine-generator load operation, and, secondly, for operation of the unit in the speed

control mode when the generator is not connected to the electrical system. The fault-tree diagrams NC 5.03a and NC 5.04a show the failure paths causing an overspeed event. All exclusive failure paths leading to a >120% speed event have been taken into account, therefore, the computed total failure probability considers all component failure combinations which could lead to this overspeed event.

The normal load operation mode is analyzed in diagram NC 5.03a. The unit is on line and as long as the generator is connected to the electric system, there is no failure of the overspeed prevention system which could lead to an overspeed event. However the overspeed prevention system must be continuously ready, and in an event of generator disconnection (load rejection) it must act to avoid an overspeed by closing the control and/or stop valves.

The second operation mode is shown in diagram NC 5.04a. This mode exists primarily during start-up and synchronizing of the unit, but also could exist during shutdown. In this mode, the turbine-generator speed is controlled by the electro-hydraulic speed control, with the mechanical-hydraulic control as back up. A failure of the control system could lead to an overspeed event by accidentally opening or holding open the control valves. In this case, the speed control system causes an overspeed, and if the overspeed prevention system with the already failed speed control cannot stop the speed increase, a >120% speed failure could happen. The fault tree for this mode of operation has one branch which initiates the overspeed event and a second branch which prevents the overspeed event.



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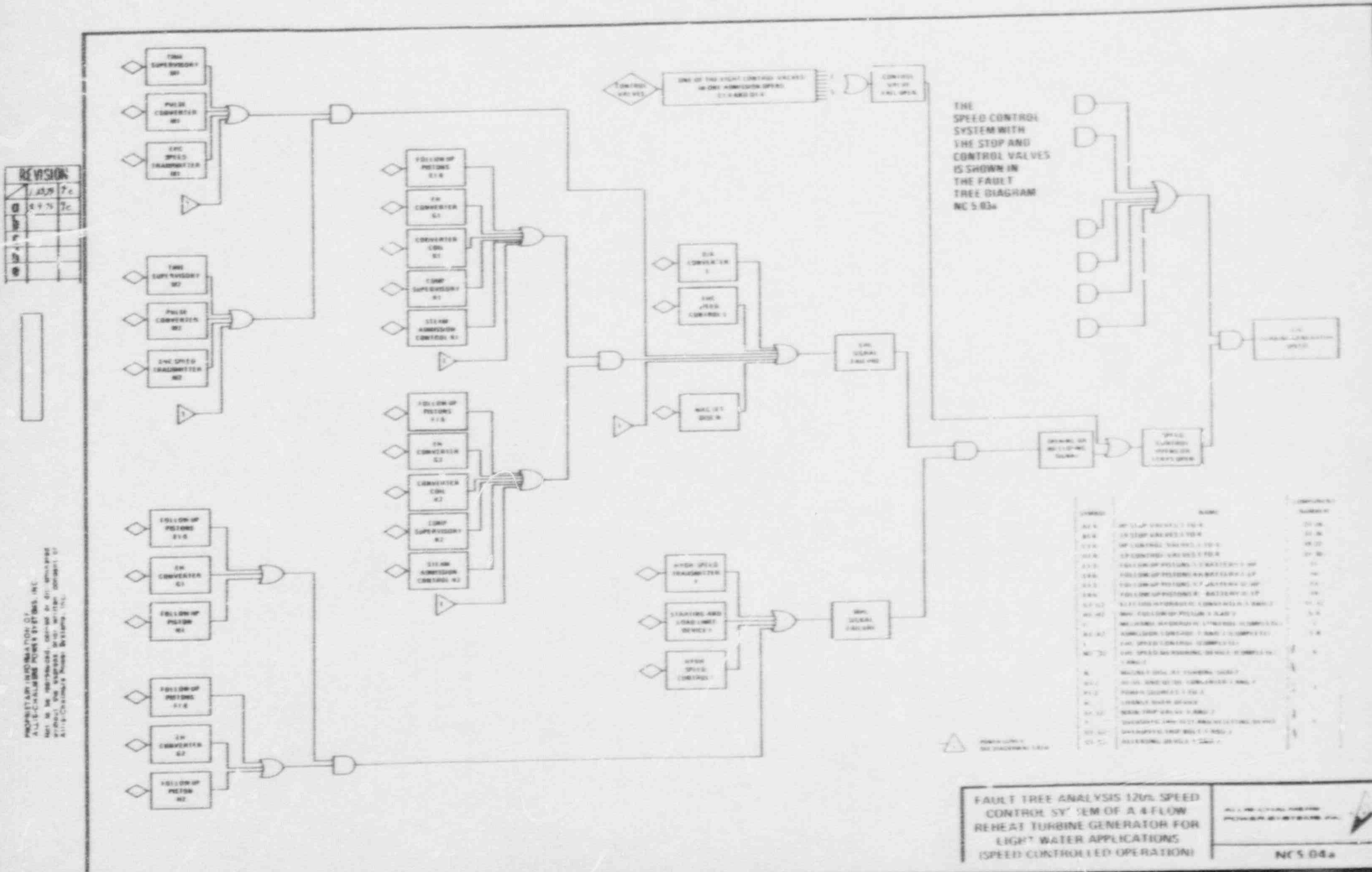

FAULT TREE ANALYSIS 120% SPEED CONTROL SY' 'EM OF A 4-FLOW REHEAT TURBINE GENERATOR FOR LIGHT WATER APPLICATIONS (SPEED CONTROLLED OPERATION)

NC5.04a

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The overspeed prevention part of the fault tree is basically the same as shown for the load rejection event shown in diagram NC 5.03a on page 82, with the trip failure and speed control failure as final results.

The >120% speed failure probability was calculated based on the two fault trees with a computer program written by KWU. The first results calculated with this program were doublechecked by Interatom* independently and with a different computer program. The Interatom results showed full agreement with the failure probability numbers calculated by KWU.

The calculated results for the load operation mode are 0.8040×10^{-7} per unit-year for a 4-flow turbine and 0.6025×10^{-7} per unit-year for a 6-flow turbine. These are the probabilities of a failure when the speed control system is required to prevent an overspeed following load rejection. This calculation includes the conservative assumption that over a 40-year lifetime of the unit there will be 40 load rejections (one per year) requiring proper operation of the overspeed prevention system. This does not include transient load disturbances which do not require operation of the overspeed prevention system.

Also, it is important to note that normal turbine trip or shutdown does not require functioning of the overspeed prevention system because the generator will only be disconnected after the turbine trip valves

* Interatom is a wholly owned subsidiary of KWU engaged in research, development and studies in the nuclear power field.

are closed. This function is normally performed by a reverse power relay which provides a very high degree of assurance that the turbine valves are shut before disconnecting the generator. The remote possibility that the unit will be improperly disconnected can reasonably be included in the above 40 events over the 40 year's lifetime.

The calculated results for the speed-control operation mode are 0.8216×10^{-7} per unit-year for a 4-flow turbine and 0.6158×10^{-7} per unit-year for a 6-flow turbine. These are the probabilities of a >120% speed failure based on the conservative assumption that the turbine-generator is operating disconnected from the network more than 336 hours per year. With this assumption, any failure of the speed control system at any time could cause an overspeed event, if the speed control would not prevent the runaway of the turbine-generator to a >120% speed failure. In case the unit operates less than 336 hours per year speed controlled and not connected to the system, this probability would be decreased.

Despite the fact that in actual practice the turbine-generator runs mostly with the generator synchronized, and that before operation with disconnected generator (start-up), a testing of the overspeed trip device and the stop and control valves can be assumed, we took the conservative assumption that the turbine-generator operates for more than 336 hours per year disconnected and over the entire year connected to the network, and added up the two calculated results to arrive at a total of 1.6256×10^{-7} per year for the probability of a >120% speed failure of a 4-flow turbine and 1.2183×10^{-7} per year of a 6-flow turbine.

The different results for the turbine-generators with 4-flow LP turbines and 6-flow LP turbines are caused by the following two deviations of these units:

a.) The 6-flow units with three instead of two LP turbines have two more LP admissions with two additional LP stop and two additional LP control valves. The LP turbine admissions with a stop and control valve in series are for a 4-flow LP turbine a "4 times 1-out-of-2-system" and for a 6-flow LP turbine a "6 times 1-out-of-2 system."

b.) For the 4-flow units with a much smaller moment of inertia, there is a probability to produce a speed slightly above 120% in case two HP control valves fail and allow a reverse stream of steam out of the two MSR's reheater tubes and end chambers into the HP turbine. Even such an event would not produce an overspeed leading to a turbine-generator missile. We took the conservative approach and added this event as a >120% speed probability for the 4-flow turbine.

The preceding calculations do not include the turbine extraction system because this system is designed and furnished by others. As shown in the reliability block diagram NC 5.01a, page 36, our equipment provides the positive closing signal for the controlled extraction valves formed out of the highly reliable hydraulic signal from the EHC speed control, the MHC speed control and the trip system. However, it is the responsibility of others to design the extraction system and select the valves. A careful study and design of this system and use of high quality, reliable valves are recommended, especially for extractions receiving

steam from secondary sources and extractions containing large amounts of stored steam energy which could drive the generator to >120% speed. Periodic testing of valves and good maintenance practices are also recommended for the extraction system.

If these recommendations are followed, we believe the extraction system will contribute only a reasonable additional probability of failure leading to overspeed. We suggest that the extraction system should be designed to contribute not more than 50% increase to the >120% speed failure probability of the turbine-generator to control a load rejection event, i.e. 50% of 4.020×10^{-8} per unit-year for a 4-flow turbine and 3.013×10^{-8} per unit-year for a 6-flow turbine

Adding these values to the previous totals of 1.6256×10^{-7} and 1.2183×10^{-7} yields the final results of $2.0276 \times 10^{-7} \approx 2.1 \times 10^{-7}$ per unit-year for the 4-flow turbine and $1.5196 \times 10^{-7} \approx 1.6 \times 10^{-7}$ per unit-year for the 6-flow turbine as the probabilities of >120% speed failures including the extraction system.

4.5 <u>Common Mode Failure</u>

The foregoing analysis is for random failures in which components are assumed to be subject to failure as a function of time at a rate following the exponential distribution in accordance with the commonly known theory of reliability analysis. In addition, it is appropriate to safety analysis to consider the so-called "common-mode" failures which may be defined as simultaneous failures of more than one component due to some common design error, improper operation or maintenance, or influence of some external condition. A pertinent example is the possibility of failure of

many components in the speed control system due to a fire in the electrical cabinet housing. Due to the nature of this type of failure, an objective and accurate quantitative estimate of common-mode failure probability is not possible. However, we have done a qualitative analysis which is summarized below under the headings of:

1. Normal external power plant environment
2. Operation and maintenance errors
3. External events.

The Common Mode Failure Chart gives the results of this study.

4.5.1 Normal External Power Plant Environment

The only significant influence of the normal external power plant environment are water and electrical interference which could adversely affect the electrical part of the speed control. Because the electrical speed control is only a part of the overall speed control system, and is hydraulically connected to the electro-hydraulic converters in parallel with the overspeed trip system and the mechanical-hydraulic control in a one-out-of-three system, it can be assumed that a common failure due to normal external power plant environment will not change the basic failure probability value due to random failures.

4.5.2 Operation and Maintenance Errors

The second column of the chart shows the possible common mode failures

COMMON MODE FAILURE CHART[1]

REASONS AND EVENTS FOR COMMON MODE FAILURES	NORMAL EXTERNAL ENVIRONMENT						OPERATION & MAINTENANCE ERRORS				EXTERNAL EVENTS			
	DIRT	TEMPERATURE	MOISTURE	VIBRATION	WATER	ELECTRICAL INTERFERENCE	MISCALIBRATION	IMPROPER TESTING	CARELESS MAINTENANCE	OPERATOR ERROR	TORNADO	FIRE	FLOOD	EARTHQUAKE
STOP & CONTROL VALVES									X					
SPEED CONTROL SYSTEM					X	X	X		X	X		X	X	
OVERSPEED TRIP SYSTEM							X	X	X	X				

X Potential for Common Mode Failure

Common Mode Failures which could lead to a > 120% speed event

due to operation and maintenance errors. In this area many different common mode failures are feasible. We believe the worst case would be careless maintenance of the entire system of stop and control valves, speed control and overspeed trip.

In this connection, it should be noted that there are no reported cases of common mode failures due to misoperation or improper maintenance in all of our historical data. We believe that with current and future units with modern operation and maintenance practices, and considering the excellent testability of the stop and control valves and overspeed trip system, the probability of such common failures should approach zero. Automatic testing of the stop and control valves, as well as the overspeed trip system after maintenance work and before each start-up clearly will reveal any maintenance error or miscalibration.

Improper testing and operator errors are eliminated by the very secure design of the ATT and the local overspeed trip test and resetting device as described previously under Elements R and T, paragraphs 4.3.8 and 4.3.10. Even if an operation error affects the speed control of both the EHC and the MHC, the unit is still protected by the trip system which is not subject to operator error. An operator error or improper testing of the stop and control valves is not possible, and as described previously, is very unlikely for the overspeed trip system. Therefore, we believe the probability of common mode failure due to operation and

maintenance errors will not significantly increase the calculated random failure probability.

4.5.3 External Events

External events could influence the electrical portion of the speed control system due to fire or flood, but as in the case of environmental effects this should not be counted as a common failure of the complete overspeed prevention system.

For all other external events and components it can be safely assumed that if the system fails, it will fail in the safe (stop and/or control valve closing) direction.

4.5.4 Conclusion

Although we know that the failure probability due to common mode failures is not zero, we believe that common mode failures affecting the whole overspeed prevention system are sufficiently unlikely that they should not change our very conservative value of failure probability calculated for random failures.

LIST OF REFERENCES

(1) Allis-Chalmers Power Systems, Inc. Report No. ER-503, July 1975 — "Turbine Missile Analysis for 1800 r/min Steam Turbine-Generators with 44" LSBs for Nuclear Power Plant Applications."

(2) S. H. Bush — "Probability of Damage to Nuclear Components Due to Turbine Failure", Nuclear Safety Vol. 14 No. 3 May-June 1973.

(3) H. D. Emmert — "Investigation of Large Steam Turbine Spindle Failure", Presented at 1955 Annual Meeting of ASME as Paper No. 55-A-172.

(4) H. Wolf and D. Sauer — "New Experimental Technique to Determine Residual Stresses in Large Turbine-Generator Components", Paper Presented at the 1974 American Power Conference, Chicago, Illinois.

(5) R. Schinn and U. Schieferstein — "Forgings from Gigantic Ingot with 140" Diameter and 881,000 lbs. Weight", Part 2, "Operational Stresses and Acceptance Criteria", Paper Presented at the 1972 International Forgemasters Meeting, Cherry Hill, New Jersey.

(6) U. Schieferstein and W. Wiemann — "Influence of Material-Inhomogeneities on Some Application Properties of Steel for Heavy Forgings", Paper Presented at the 1975 International Forgemasters Meeting, Paris, April 20-25, 1975.

(7) R. Schinn — "Quality Control of Large Castings and Forgings for Steam Turbine-Generators", 1971 American Power Conference, Proceedings, Vol. 33.

(8) Allis-Chalmers Power Systems, Inc. Engineering Report #E415, November 1974 — "Speed Control of 1800 r/min Steam Turbine-Generators for Light Water Reactor Applications."

(9) IEEE Std 352-1972/ ANSI N41.4 — IEEE Trial-Use Guide: "General Principles for Reliability Analysis of Nuclear Power Generating Station Systems."

(10) Igor Bazovsky — "Reliability Theory and Practice."

(11) B. J. Garrick, et al — "HN 190 Relibility Analysis of Nuclear Power Plant Protective Systems."

ENCLOSURE 5 TO TXX-92486

Supplement 6 to NUREG 0797, Safety Evaluation Report related to operation of CPSES Units 1 and 2, November 1984
Table 10.1 (page 10-9)

Table 10.1 Turbine system reliability criteria

Probability, yr^{-1}		
Favorably oriented turbine	Unfavorably oriented turbine	Required license action
(1) $P_1 < 10^{-4}$	$P_1 < 10^{-5}$	This is the general, minimum reliability requirement for loading the turbine and bringing the system on line.
(2) $10^{-4} < P_1 < 10^{-3}$	$10^{-5} < P_1 < 10^{-4}$	If this condition is reached during operation, the turbine may be kept in service until the next scheduled outage, at which time the licensee is to take action to reduce P_1 to meet the appropriate criterion in item (1) above before returning the turbine to service.
(3) $10^{-3} < P_1 < 10^{-2}$	$10^{-4} < P_1 < 10^{-3}$	If this cond[illegible]hed during operation, the turbine is to be isolated from the steam supply within 60 days, at which time the licensee is to take action to reduce P_1 to meet the appropriate criterion in Item (1) above before returning the turbine to service.
(4) $10^{-2} < P_1$	$10^{-3} < P_1$	If this condition is reached at any time during operation, the turbine is to be isolated from the steam supply within 6 days, at which time the licensee is to take action to reduce P_1 to meet the appropriate criterion in Item (1) above before returning the turbine to service.

ENCLOSURE 6 TO TXX-92486

Federal Register Vol. 51, No. 44, Rules and Regulations, March 6, 1986, page 7751

§ 50.21(b) or § 50.22 or for a testing facility will likely be found to involve significant hazards considerations, if operation of the facility in accordance with the proposed amendment involves one or more of the following:

(i) A significant relaxation of the criteria used to establish safety limits.

(ii) A significant relaxation of the bases for limiting safety system settings or limiting conditions for operation.

(iii) A significant relaxation in limiting conditions for operation not accompanied by compensatory changes, conditions, or actions that maintain a commensurate level of safety (such as allowing a plant to operate at full power during a period in which one or more safety systems are not operable).

(iv) Renewal of an operating license.

(v) For a nuclear power plant, an increase in authorized maximum core power level.

(vi) A change to technical specifications or other NRC approval involving a significant unreviewed safety question.

(vii) A change in plant operation designed to improve safety but which, due to other factors, in fact allows plant operation with safety margins significantly reduced from those believed to have been present when the license was issued. *Id.*

e. *Examples of Amendments That Are Considered Not Likely To Involve Significant Hazards Considerations Are Listed Below.* The statement of considerations for the interim final rules listed the following examples of amendments the Commission considered not likely to involve significant hazards considerations. 48 FR 14869. It explained that unless the specific circumstances of a license amendment request lead to a contrary conclusion when measured against the standards in § 50.92, then, pursuant to the procedures in § 50.91, a proposed amendment to an operating license for a facility licensed under § 50.21(b) or § 50.22 or for a testing facility will likely be found to involve no significant hazards considerations, if operation of the facility in accordance with the proposed amendment involves only one or more of the following:

(i) A purely administrative change to technical specifications: for example, a change to achieve consistency throughout the technical specifications, correction of an error, or a change in nomenclature.

(ii) A change that constitutes an additional limitation, restriction, or control not presently included in the technical specifications, *e.g.*, a more stringent surveillance requirement.

(iii) For a nuclear power reactor, a change resulting from a nuclear reactor core reloading, if no fuel assemblies significantly different from those found previously acceptable to the NRC for a previous core at the facility in question are involved. This assumes that no significant changes are made to the acceptance criteria for the technical specifications, that the analytical methods used to demonstrate conformance with the technical specifications and regulations are not significantly changed, and that NRC has previously found such methods acceptable.

(iv) A relief granted upon demonstration of acceptable operation from an operating restriction that was imposed because acceptable operation was not yet demonstrated. This assumes that the operating restriction and the criteria to be applied to a request for relief have been established in a prior review and that it is justified in a satisfactory way that the criteria have been met.

(v) Upon satisfactory completion of construction in connection with an operating facility, a relief granted from an operating restriction that was imposed because the construction was not yet completed satisfactorily. This is intended to involve only restrictions where it is justified that construction has been completed satisfactorily.

(vi) A change which either may result in some increase to the probability or consequences of a previously-analyzed accident or may reduce in some way a safety margin, but where the results of the change are clearly within all acceptable criteria with respect to the system or component specified in the Standard Review Plan, *e.g.*, a change resulting from the application of a small refinement of a previously used calculational model or design method.

(vii) A change to conform a license to changes in the regulations, where the license change results in very minor changes to facility operations clearly in keeping with the regulations.

(viii) A change to a license to reflect a minor adjustment in ownership shares among co-owners already shown in the license. *Id.*

[As discussed below, the Commission has added examples (ix) and (x) in response to comments on the interim final rules.]

(ix) A repair or replacement of a major component or system important to safety, if the following conditions are met:

(1) The repair or replacement process involves practices which have been successfully implemented at least once on similar components or systems elsewhere in the nuclear industry or in other industries, and does not involve a significant increase in the probability or consequences of an accident previously evaluated or create the possibility of a new or different kind of accident from any accident previously evaluated; and

(2) The repaired or replacement component or system does not result in a significant change in its safety function or a significant reduction in any safety limit (or limiting condition of operation) associated with the component or system.

(x) An expansion of the storage capacity of a spent fuel pool when all of the following are satisfied:

(1) The storage expansion method consists of either replacing existing racks with a design which allows closer spacing between stored spent fuel assemblies or placing additional racks of the original design on the pool floor if space permits;

(2) The storage expansion method does not involve rod consolidation or double tiering;

(3) The Keff of the pool is maintained less than or equal to 0.95; and

(4) No new technology or unproven technology is utilized in either the construction process or the analytical techniques necessary to justify the expansion.

II. Responses to Comments on Interim Final Rules

The comments are described in somewhat greater detail in an attachment to SECY-85-209A.

A. Clarity of Standards

1.1 Comments—A group of commenters state that the three standards in § 50.92(c) are unclear and argue that the examples in the statement of considerations—which they believe are clearer than the standards—should be made part of the rule; otherwise, they argue, the examples have no legal significance.

Response—The Commission disagrees with the request. As explained in response to the comments on the proposed rule (*see* 48 FR 14864), the commenters are correct that the examples have no binding legal significance. However, they do provide guidance to the staff, licensees and to the general public about the way the standards may be interpreted by the Commission. The Commission did consider combining the standards and examples as a single set of criteria in the interim final rules, but decided against this because (i) the standards and examples had proved useful over time, (ii) the staff had used all three standards