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5D TURBINE GENERATOR MISSILES

CONSIDERATION OF THE CONSEQUENCES OF A TURBINE GENERATOR FAILURE IN WHICH MISSILES ARE GENERATED

INTRODUCTION AND CONCLUSIONS

The present advanced state of the art of rotor forgings and inspection techniques guarantee, for all practical purposes, defect-free turbine rotors. Furthermore, Westinghouse conservative designs eliminate any harmful stress-concentration points. This is confirmed by the record of Westinghouse turbine-generators - no unit has ever experienced a rotor or disc failure.

Hence the only envisaged cause that might lead to a turbine-generator failure is excessive overspeeding.

Due to the redundancy and reliability of the turbine control and protection system and of the steam system, the occurrence of a unit overspeeding above the design value, i.e. 115%, is very remote.

However, the consequences of a turbine-generator runaway, caused by all the steam admission valves stuck fully open upon a full load rejection, have been evaluated, for purpose of analysis.

As it will be shown later, missiles will be generated only from the rotating parts of the low-pressure turbine, but they will not have sufficient energy to penetrate through the turbine casings.

GENERAL DESCRIPTION OF THE TURBINE UNITS

HIGH PRESSURE TURBINE

The high pressure turbine element, shown in Figure 1 is of a double flow design; therefore, it is inherently thrust balanced. Steam from the four control valves enters at the center of the turbine element through four inlet pipes, two in the base and two in the cover. These pipes feed four double flow nozzle chambers flexibly connected to the turbine casing. Each nozzle chamber is free to expand and contract relative to the adjacent chambers.

Steam leaving the nozzle chambers passes through the rateau control stages and flows through the reaction blading. The reaction blading is mounted in blade rings shown in Figure 2, which in turn are mounted in the turbine casing. The blade rings are centerline supported to insure center alignment while allowing for differential expansion between the blade ring and the casing. This design reduces casing thermal distortion and thus, seal clearances are more readily maintained.

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Steam exhausts from the high pressure turbine base, through cross-under piping, to the four combined moisture separator live steam reheater assemblies.

The high-pressure rotor is made of NiCrMoV alloy steel. The specified minimum mechanical properties are as follows:

Tensile Strength, psi, min.	100,000
Yield Strength, psi, min. (0.2% offset)	80,000
Elongation in 2 inches, per cent, min.	18
Reduction of Area, per cent, min.	45
Impact Strength, Charpy V-Notch, ft-lb (min. at room temperature)	60
50% Fracture Appearance Transition Temperature, °F, max.	50

The main body of the rotor weighs approximately 100,000 lb. The approximate values of the transverse centerline diameter, the maximum diameter, and the main body length are 36", 66" and 138" respectively.

The blade rings and the casing cover and base are made of carbon steel castings. The specified minimum mechanical properties are as follows:

Tensile Strength, psi, min.	70,000
Yield Strength, psi, min.	36,000
Elongation in 2", per cent, min.	22
Reduction of Area, per cent, min.	35

The bend test specimen shall be capable of being bent cold through an angle of 90 degrees and around a pin one inch in diameter without cracking on the outside of the bent portion.

The approximate weights of the four blade rings, the casing cover, and the casing base are 80,000 lb., 115,000 lb., and 115,000 lb., respectively.

The casing cover and base are tied together by means of more than 100 studs. The stud material is an alloy steel having the following mechanical properties:

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	Size, Inches		
	2-1/2 and less	Over 2-1/2 to 4 inch	Over 4 to 7 inch
Tensile Strength, psi, min.	125,000	115,000	110,000
Yield Strength, psi, min. (0.2% offset)	105,000	95,000	85,000
Elongation in 2 inches, per cent, min.	16	16	16
Reduction of Area, per cent, min.	50	50	45

The studs have length ranging from 17 to 66 inches and diameter ranging from 2.75" to 4.5". About 90% of them have diameter ranging between 2.5 and 4 inches. The total stud cross-sectional area is about 900 in^2 and the total stud free-length volume is about 36,000 in^3.

LOW PRESSURE TURBINE

The double flow low pressure turbine, shown in Figure 3, incorporates high efficiency blading, diffuser type exhaust and liberal exhaust hood design. The low pressure turbine cylinders are fabricated from steel plate to provide uniform wall thickness thus reducing thermal distortion to a minimum. The entire outer casing is subjected to low temperature exhaust steam.

The temperature drop of the steam from its inlet to the L.P. to its exhaust from the last rotating blades is taken across three walls; an inner cylinder number 1, a thermal shield, and an inner cylinder number 2, as shown in Figure 4. This precludes a large temperature drop across any one wall except the thermal shield which is not a structural element, thereby virtually eliminating thermal distortion. The fabricated inner cylinder number 2, is supported by the outer casing at the horizontal centerline and is fixed transversely at the top and bottom and axially at the centerline of the steam inlets, thus allowing freedom of expansion independent of the outer casing. Inner cylinder number 1 is, in turn, supported by inner cylinder number 2, at the horizontal centerline and fixed transversely at the top and bottom and axially at the centerline of the steam inlets, thus allowing freedom of expansion independent of inner cylinder number 2. Inner cylinder number 1 is surrounded by the thermal shield.

The steam leaving the last row of blades flows into the diffuser where the velocity energy is converted to pressure energy, thus improving efficiency and reducing the excitation forces on the last rotating row of blades.

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The low pressure rotors are made of NiCrMoV alloy steel. The specified minimum mechanical properties are as follows:

Tensile Strength, psi, min.	115,000
Yield Strength, psi, min. (0.2% offset)	100,000
Elongation in 2 inches, per cent, min.	16
Reduction of Area, per cent, min.	40
Impact Strength, Charpy V-Notch, ft-lb min. at room temp.	40
50% Fracture Appearance Transition Temperature, °F, max.	80

The shrunk-on discs are made of NiCrMoV alloy steel. There are twelve discs shrunk on the shaft with six per flow. These discs experience different degrees of stress when in operation. The present design shows that disc No. 3, starting from the transverse centerline, experiences the highest stress, while disc No. 6 experiences the lowest. The minimum specified mechanical properaties for the discs are as follows:

	Disc No. 3	Disc No. 1,2,4,5 and 6
Tensile Strength, psi, min.	130,000	120,000
Yield Strength, psi, (0.2% offset)	120,000-135,000	110,000-125,000
Elongation in 2" (Disc Hub), per cent, min.	14	15
Elongation in 2" (Disc Rim), per cent, min.	16	17
Reduction of Area (Disc Hub), per cent, min.	35	38
Reduction of Area (Disc Rim), per cent, min.	40	43
Impact Strength, (Hub and Rim), Charpy V-notch, ft lb min. at room temp.	50	50
50% Fracture Appearance Transition Temperature (Disc Hub and Rim) °F, max.	0	0

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The outer cylinder and the two inner cylinders are mainly made of ASTM A-285 Grade C material. The minimum specified properties are as follows:

Tensile Strength, psi, min.	55,000 psi
Yield Strength, psi, min.	30,000 psi
Elongation in 8", per cent, min.	24
Elongation in 2", per cent, min.	28

Whenever plates of thickness > 2" are employed, they are made of ASTM A-212 Grade A.

CONSEQUENCES OF TURBINE-GENERATOR UNIT OVERSPEEDING

LOW PRESSURE TURBINE

Experience and tests have shown that the mode of failure of a disc, should it occur, is mainly rupture in two or four parts. The broken parts would then be ejected normally to the rotation axis. Hence, the potential missiles considered for purposes of analysis are:

a) Half disc

b) A quarter of disc

There are twelve discs shrunk on each low-pressure turbine rotor, with six discs per flow. Numbering the discs from the steam admission, discs No. 1, 2 and 3 are contained within the inner cylinder No. 1, the inner cylinder No. 2, and the outer cylinder (reference is made to Figures 3 and 4). Therefore, if one of these discs breaks, it has to go through the corresponding stationary blade ring, the inner cylinder No. 1, the inner cylinder No. 2, and the outer cylinder. Discs No. 4 and 5 are contained within the inner cylinder No. 2 and the outer cylinder. Hence, if one of these fails, it has to pass through the directly opposite blade ring, the inner cylinder No. 2 and the outer cylinder. Disc No. 6 is partially contained within the inner cylinder No. 2 and partially within the diffuser and within the outer cylinder. If parts of this disc come loose, they have to go through the directly opposite blade ring and the outer cylinder.

The thickness of the back plate of the three cylinders is as follows:

Inner cylinder No. 1	=	2 inches
Inner cylinder No. 2	=	1.25 inches
Outer cylinder	=	1.25 inches

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The bursting speed of each disc has been calculated with a stress analysis based on a tensile strength 20 per cent higher than the minimum specified tensile strength. The 20 per cent increase conservatively account for the actual value of the tensile strength, usually observed to be higher than the minimum specified. The values of the minimum and maximum bursting speeds of each disc are listed in the following table

TABLE 1

DISC BURSTING SPEED

Type of Disc	Bursting Speed (per cent of nominal)	
	Maximum	Minimum
Disc No. 1	179	163
Disc No. 2	181	165
Disc No. 3	175	153
Disc No. 4	179	163
Disc No. 5	178	162
Disc No. 6	187	171

As the above table shows, the maximum speed at which the unit might run with no disc failure is 175% of nominal. At this speed, disc No. 3 will burst. As one of the first discs ruptures, the steam flow between the blades of the remaining discs is significantly reduced, the turbine-generator is slowed down, and further disc failures are not anticipated. Since the actual value of the bursting speed of each disc will be between the maximum and minimum previously mentioned, the potentiality of bursting each one of the first five discs exists. The probability of disc No. 6 bursting is more remote. The consequences of rupture of any one of these discs at the maximum speed that the unit might approach in case of turbine runaway have been evaluated and the results are summarized in the following pages.

Table 2 lists the values of the rim radius, the weight, the ejection velocity and ejection translational energy of each disc quarter, at 175% of nominal speed. Table 3 lists the same parameters for half discs.

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TABLE 2

RUPTURE IN FOUR QUARTERS AT 175% OF NOMINAL SPEED

Type of Disc	Rim Radius (inches)	Weight (lb)	Ejection Velocity (ft/sec)	Ejection Translation Kinetic Energy (ftxlb)
Quarter of Disc No. 1	51.875	2050	855	23.3×10^6
Quarter of Disc No. 2	51.875	1912.5	855	21.7×10^6
Quarter of Disc No. 3	51.875	2455	855	25.0×10^6
Quarter of Disc No. 4	51.234	2575	845	28.6×10^6
Quarter of Disc No. 5	49.162	2900	810	29.6×10^6
Quarter of Disc No. 6	43.800	3100	722	25.1×10^6

TABLE 3

RUPTURE IN TWO HALVES AT 175% OF NOMINAL SPEED

Type of Disc	Rim Radius (inches)	Weight (lb)	Ejection Velocity (ft/sec)	Ejection Translational Kinetic Energy (ftxlb)
Half of Disc No. 1	51.875	4100	605	23.3×10^6
Half of Disc No. 2	51.875	3825	605	21.7×10^6
Half of Disc No. 3	51.875	4910	605	25.0×10^6
Half of Disc No. 4	51.234	5150	598	28.6×10^5
Half of Disc No. 5	49.162	5800	573	29.6×10^6
Half of Disc No. 6	43.800	6200	510	25.1×10^6

Disc No. 1, No. 2, and No. 3

Rupture of disc No. 3 has been assumed for purpose of analysis because the four quarters of this disc have more translational Kinetic energy than disc No. 1 and No. 2. As the four quarters come loose, they strike and deeply deform the inner cylinder No. 1 and cause some deformation of the inner cylinder No. 2 and of the outer cylinder of less extent.

The rupture is expected to be contained within the unit and no outside missile is anticipated to be generated.

The deformation energy per unit volume of the cylinder material has been evaluated under "static" and "dynamic" loading, based on both minimum specified and actual averaged mechanical properties. Table 4 summarizes the values of the deformation energy per unit volume up to 100%, 75%, and 50% of the total elongation, respectively.

TABLE 4

DEFORMATION ENERGY PER UNIT VOLUME

A. BASED ON THE MINIMUM SPECIFIED MECHANICAL PROPERTIES

	Up to 50% E_u	Up to 75% E_u	Up to 100% E_u
Under "static" loading	4,400 $\frac{\text{in lb}}{\text{in}^3}$	7,000 $\frac{\text{in lb}}{\text{in}^3}$	10,200 $\frac{\text{in lb}}{\text{in}^3}$
Under "dynamic" loading	7,900 $\frac{\text{in lb}}{\text{in}^3}$	11,900 $\frac{\text{in lb}}{\text{in}^3}$	15,800 $\frac{\text{in lb}}{\text{in}^3}$

B. BASED ON THE ACTUAL AVERAGED MECHANICAL PROPERTIES

	Up to 50% E_u	Up to 75% E_u	Up to 100% E_u
Under "static" loading	6,400 $\frac{\text{in lb}}{\text{in}^3}$	9,900 $\frac{\text{in lb}}{\text{in}^3}$	14,300 $\frac{\text{in lb}}{\text{in}^3}$
Under "dynamic" loading	9,000 $\frac{\text{in lb}}{\text{in}^3}$	13,500 $\frac{\text{in lb}}{\text{in}^3}$	18,000 $\frac{\text{in lb}}{\text{in}^3}$

It is expected that in order to penetrate through the inner cylinder No. 1, the ruptured disc quarters shall have the Kinetic energy necessary to deform about 1/3 of the inner cylinder No. 1 volume to between 50% and 75% of the actual total elongation, i.e., between 100×10^6 ft lb and 150×10^6 ft lb. The anticipated Kinetic energy of 4 quarters of disc No. 3 is at the lower limit of the above range, i.e., 100×10^6 ft lb.

For disc fragments to become missiles, they have to violate not only the integrity of the inner cylinder No. 1, but also that of the inner cylinder No. 2 and of the outer cylinder. As mentioned earlier, quarters of disc No. 3 are not expected to violate the integrity of inner cylinder No. 1. Should violation occur for some unknown reasons, the Kinetic energy of the quarters would be small. Therefore, for these fragments to leave the unit, they shall have enough Kinetic energy to deform a significant amount of inner cylinder No. 2 and outer cylinder, rather than just the energy necessary to perforate the back plates of these cylinders.

For these reasons, we do not expect external missiles to be generated because of failure of one of the first three discs.

Disc No. 4 and No. 5

Rupture of disc No. 5 has been conservatively assumed for purpose of analysis because the four quarters of this disc have more translational Kinetic energy than disc No. 4. As the four quarters come loose, they strike and deeply deform the inner cylinder No. 2, and cause some deformation of the outer cylinder.

The rupture is expected to be contained within the unit and no outside missile is anticipated to be generated.

It is expected that, in order to penetrate through the inner cylinder No. 2, the ruptured disc quarters shall have the Kinetic energy necessary to deform about 25% of the inner cylinder No. 2 volume to between 50% and 75% of the actual total elongation, i.e., between 136×10^6 and 200×10^6 ft lb. The anticipated Kinetic energy of 4 quarters of disc No. 5 is less than 120×10^6 ft lb.

For disc fragments to become missiles, they have to violate not only the integrity of the inner cylinder No. 2, but also that of the outer cylinder. Therefore, ejection of quarters of disc No. 4 and 5 outside the unit is not expected.

Disc No. 6

This disc is the least stressed disc, and the disc that has the highest bursting speed range, i.e., 171%-187% of nominal. The probability of reaching this speed range is quite remote, because one of the other discs is anticipated to fail at lower speed, preventing the unit from reaching the bursting speed range of disc No. 6. For purpose of analysis it is assumed that disc No. 6 bursts at the maximum running speed of 175% of nominal.

The damage caused by this failure is expected to be contained within the unit.

Upon bursting, the ejected quarters will strike the coupling flanges of the outer cylinder center and the outer cylinder side. It is expected that, in order to penetrate through the outer cylinder, the ejected quarters shall have the Kinetic energy required to deform the directly opposite blade ring, the above mentioned flanges and a two-disc-hub wide portions of the outer cylinder, for a total of

150,000 in^3, to between 50% and 75% of the actual total elongation, i.e., between 112 x 10^6 and 168 x 10^6 ft lb. Since the anticipated Kinetic energy of 4 disc No. 6 quarters, i.e., 100 x 10^6 ft lb, is below the lower limit of the required energy range, no external missile is anticipated.

HIGH PRESSURE TURBINE

Due to the very large margin between the high pressure spindle bursting speed and the maximum speed at which the steam can drive the unit with all the admission valves fully open, the probability of spindle failure is practically zero. Therefore, no harmful missile is anticipated in case of turbine runaway.

Based on the admission steam thermodynamic properties and blade geometry, the maximum theoretical speed at which the unit may run is 205% of nominal.

Based on the stress analysis of the low-pressure discs, the maximum actual speed at which the unit may run is 175% of nominal.

The minimum bursting speed of the spindle, based on the minimum specified mechanical properties of the spindle material, is 270% of nominal. The actual bursting speed is closer to 300% of nominal than 270%.

Hence, the actual margin between the bursting speed and the maximum running speed is of the order of 125% of nominal, i.e., 300%-175%.

No failure of the H.P. is anticipated as a consequence of a unit runaway; and therefore, no missiles are expected to be generated.

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Figure	Description	P.D.L. or Photograph
1	High Pressure Cylinder	PDL 1250-146
2	Blade Rings	PDL 1250-9
3	Rateau Control Stage	PDL 1250-155
4	Exploded View of 1800 Rpm L.P. Turbine	P-67040

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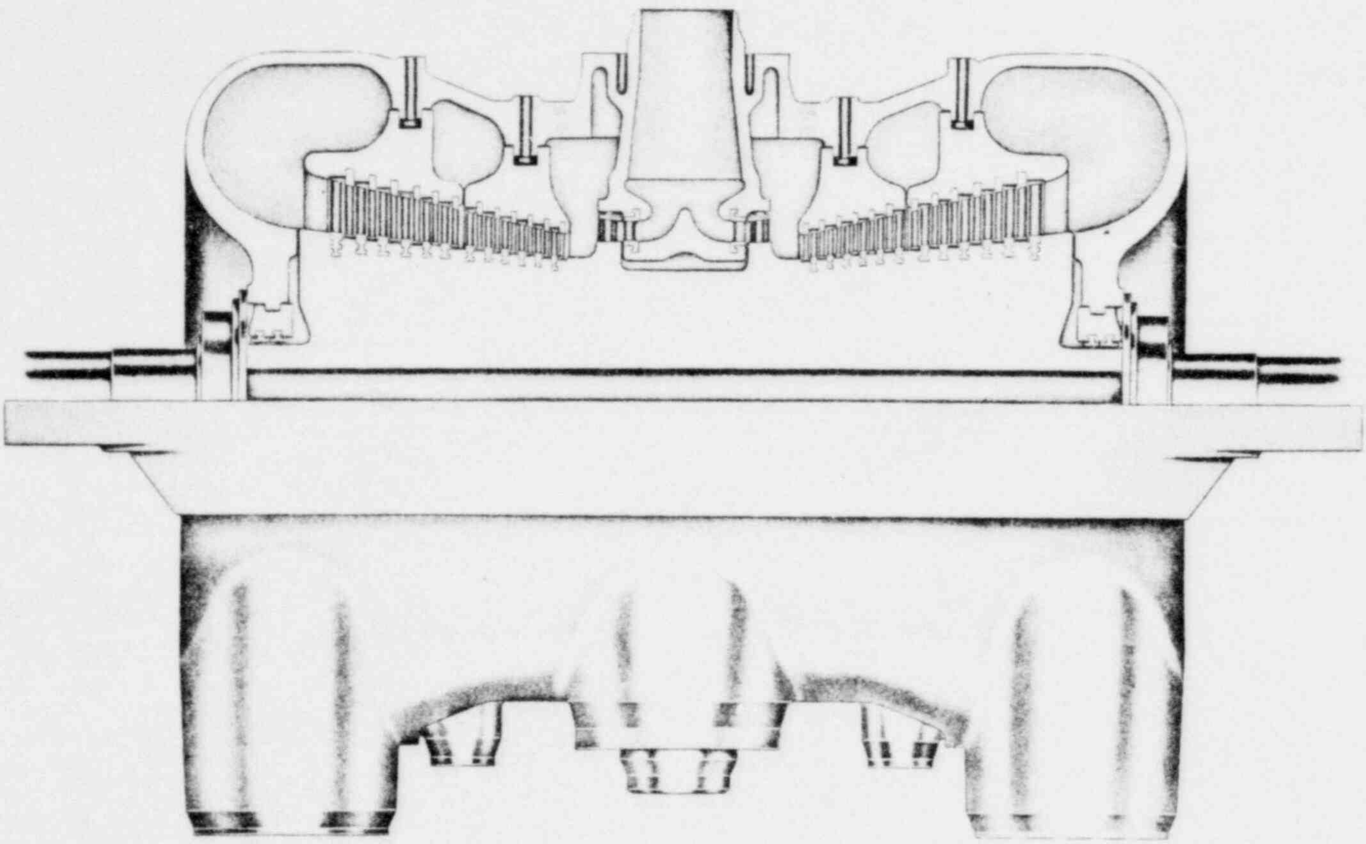
HIGH PRESSURE CYLINDER
1800 RPM DOUBLE-FLOW DESIGN



FEATURES

1. Four separate nozzle chambers permit freedom of expansion and contraction during starting and load changes.
2. Double flow design insures thrust balance.
3. Rotor checked in heater box for dynamic balance prior to shipment.
4. Ultrasonic test of rotor performed at steel mill and at the Westinghouse factory.

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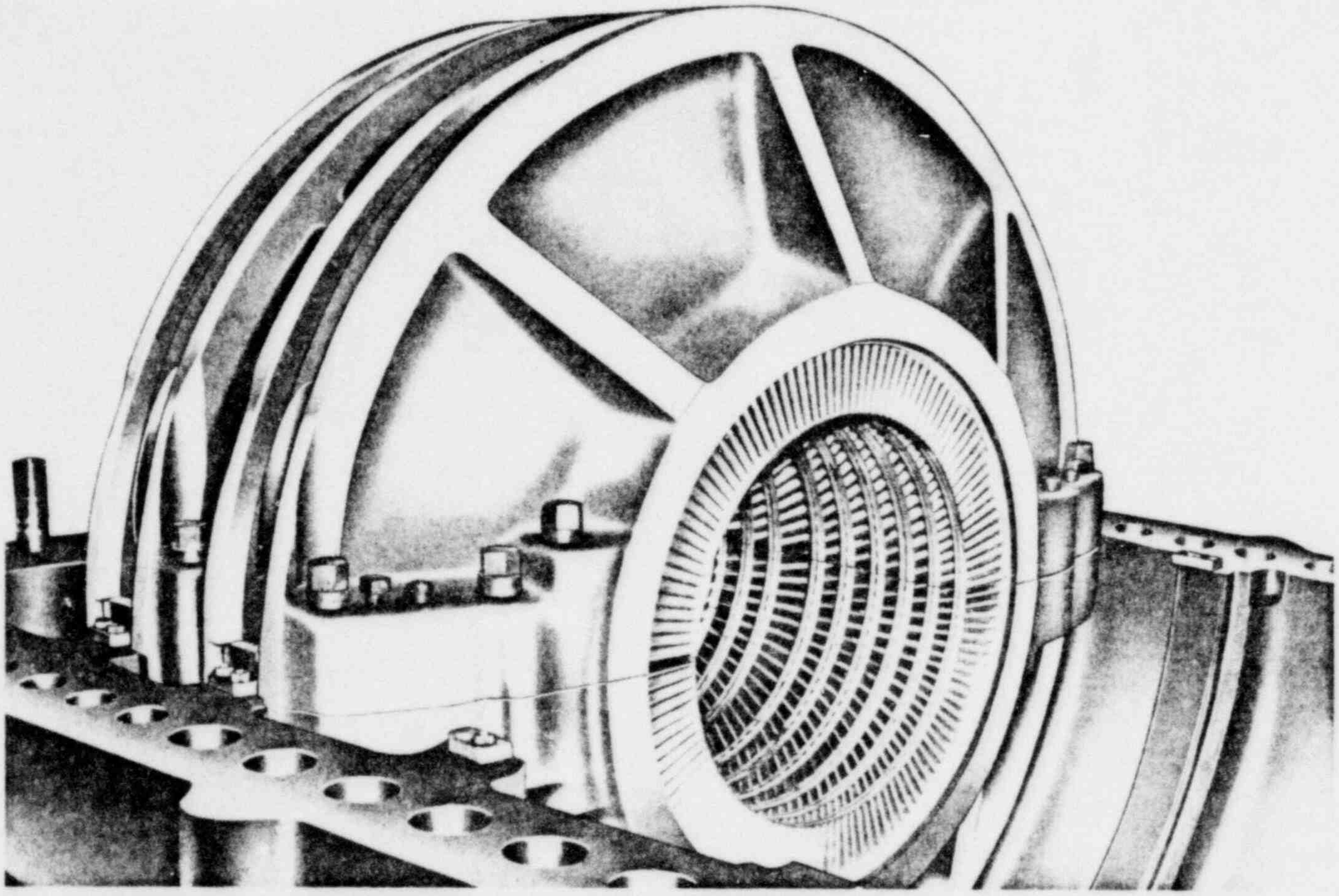
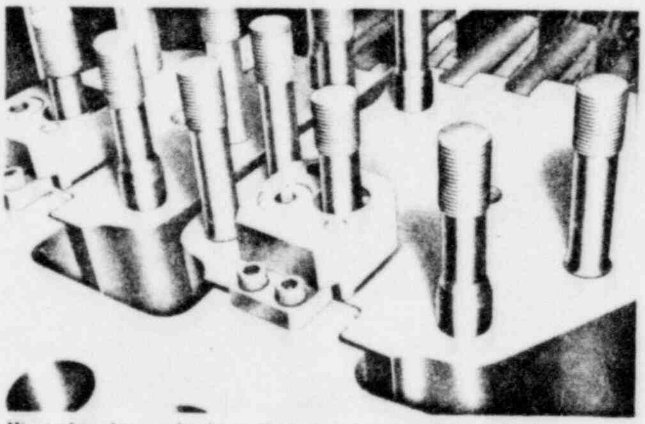
BLADE RINGS



Blade rings of large high-pressure, high temperature turbine, with stationary blades in place.

FEATURES

1. Centerline supporting block insures center alignment while allowing differential expansion between blade ring and cylinder.
2. Blades are inserted in blade ring halves.
3. Tongue and groove holds blade ring in position.
4. Metallic seals between blade rings and cylinder prevent leakage of steam in support grooves.
5. Upper plate, in cylinder cover, prevents any "riding-up" of the blade ring.



View of turbine cylinder and blade ring, showing method of supporting and locking lower blade ring in position.

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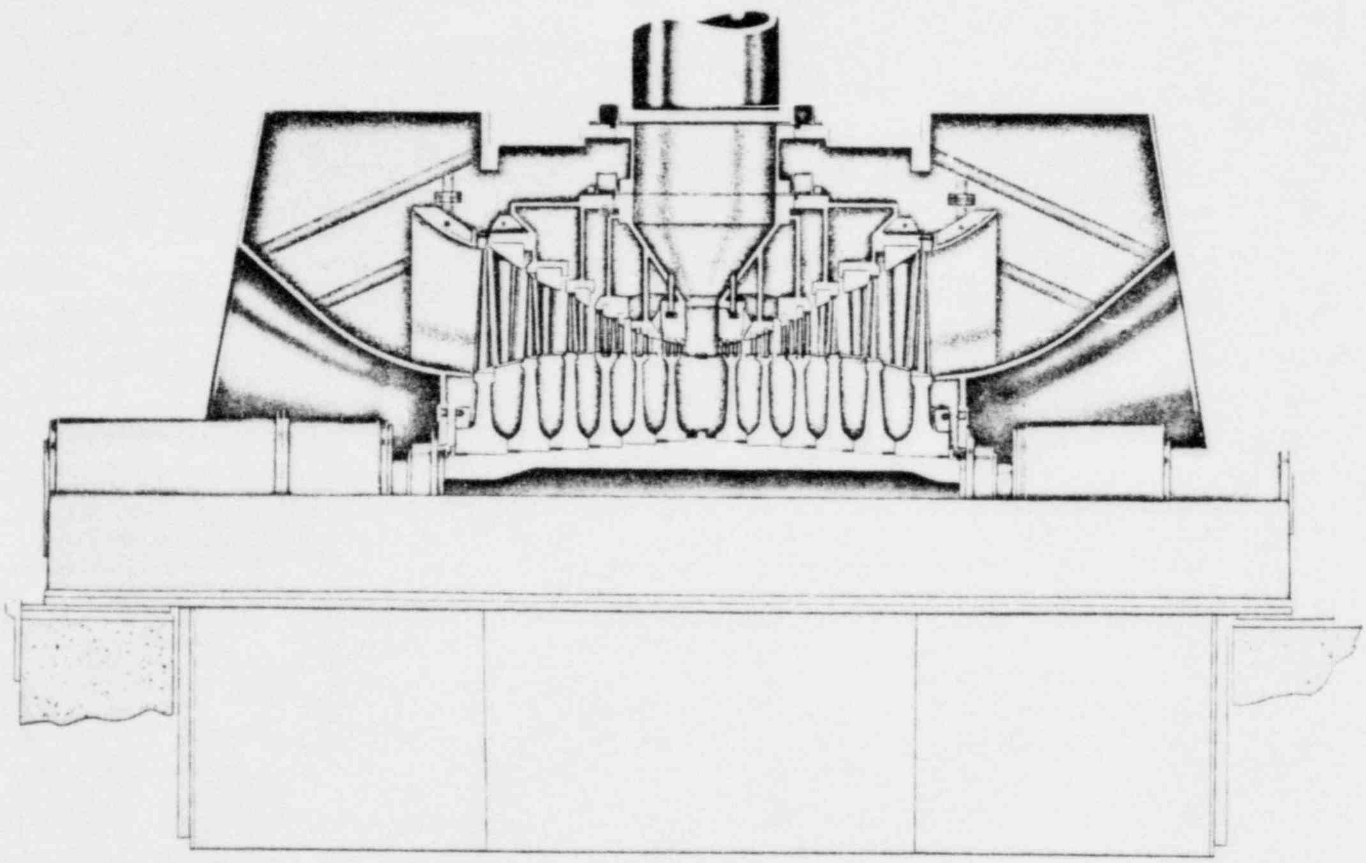
LOW-PRESSURE ELEMENT
1800-RPM DOUBLE-FLOW DESIGN



FEATURES

1. Blade ring, supported at the horizontal centerline and fixed transversely at the top and bottom by dowel pins, allows freedom of expansion independent of the casing.
2. Entire exhaust casing is at exhaust steam temperature.
3. Exhaust hood of laboratory-proved design minimizes hood loss.
4. Provision for extraction zones with moisture removal.
5. Casing and blade ring of fabricated steel construction.

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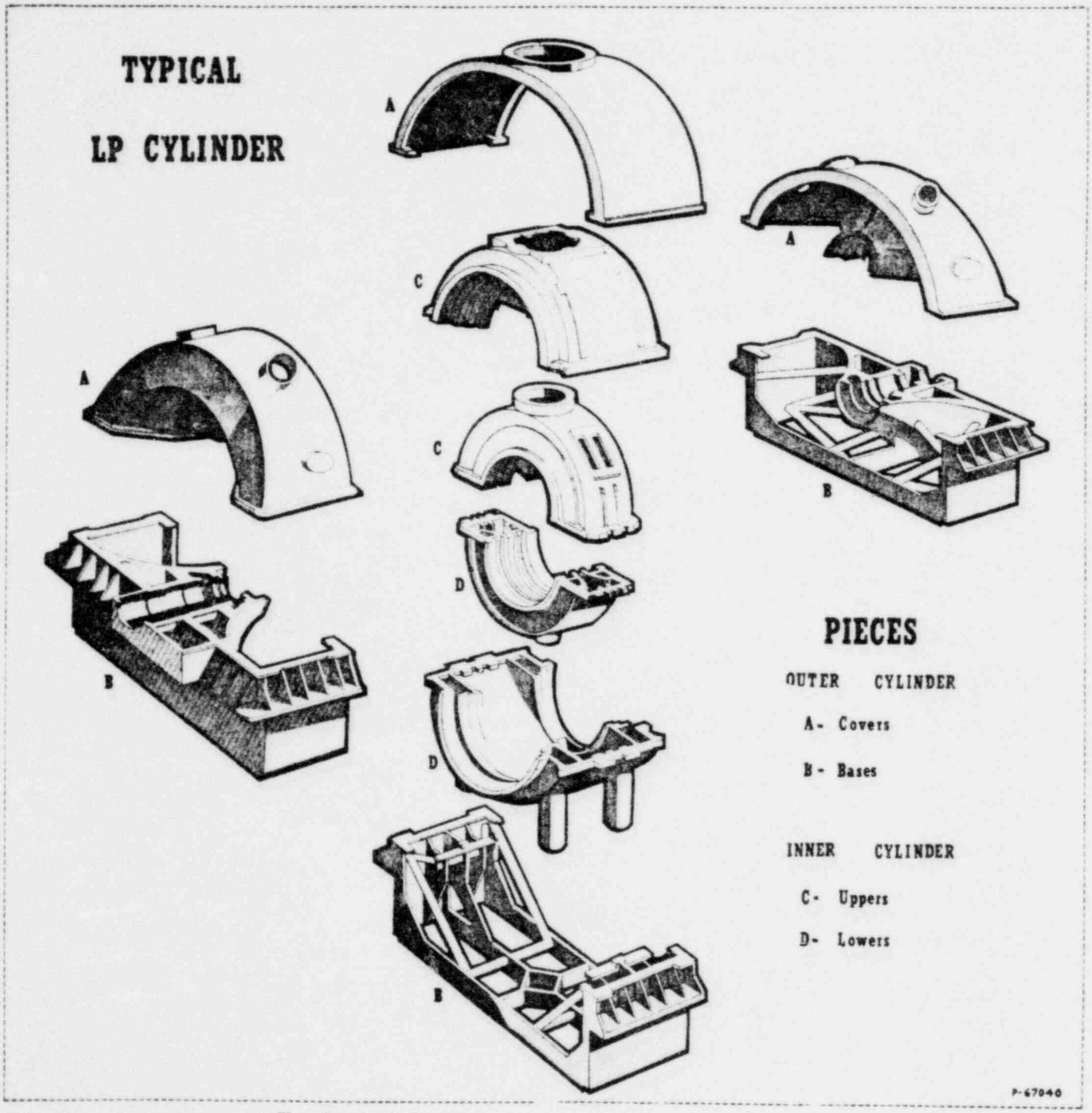

Exploded View of Low Pressure Unit
2.

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