GA-A14530
UC-77

ALTERNATE CIRCULATOR DESIGN CONCEPT NO. 1 FOR A 300 MW(e) GCFR DEMONSTRATION PLANT – TWO STAGE AXIAL FLOW HELIUM CIRCULATOR AND TWO STAGE STEAM TURBINE DRIVE

by
G. MOURITZEN

Prepared under
Contract EY-76-C-03-0167
Project Agreement No. 23
for the San Francisco Operations Office
Department of Energy

GENERAL ATOMIC PROJECT 3228
DATE PUBLISHED: DECEMBER 1977

GENERAL ATOMIC COMPANY

ABSTRACT

This report is being issued by the GCFR Primary Systems Components Branch to document the conceptual design and analysis of an alternate main helium circulator for a 300 MW(e) GCFR demonstration plant. A special feature of the design is that it is based on HTGR circulator technology. Particular emphasis was placed upon avoiding deviation from this established technology except in areas where specific requirements made such variations advisable. Two circulator stages and two turbine stages are used in this alternate design. An integrated circulator and turbine on a single shaft as well as a separate circulator unit with an external turbine drive unit were investigated. Safety requirements and accident operating conditions were investigated and a special thrust balancing system was developed to satisfy thrust load requirements for all normal and upset operating conditions. This report describes an axial flow circulator and a series flow steam turbine drive that will provide the helium flow requirements and also be compatible with the specified steam flow and temperature requirements as well as providing means for simplifying circulator test facility and hot flow reactor test programs.

ACKNOWLEDGEMENTS

Howard Wesch and Glenn Thurston made contributions to the sections on blade vibrations and accident conditions.

CONTENTS

CONTENTS (continued)

FIGURES

FIGURES (continued)

TABLES

1. INTRODUCTION

A preliminary analysis was made of the main circulator for an updated 300 MW(e) Gas Cooled Fast Breeder Reactor (GCFR) demonstration plant. The analysis was based on design requirements established by the GCFR Systems Analysis Branch for a GCFR reactor with a helium loop Δ P of 60 psi (0.413 MPa). The methods used for designing the circulators were the same as for the High-Temperature Gas-Cooled Reactor (HTGR).

This report describes the preliminary design of a main helium circulator for the 300 MW(e) GCFR demonstration plant. Three main circulators and three auxiliary circulators are to be used in the demonstration plant. The main circulator described in this report consists of an axial flow, two stage helium circulator (compressor) driven by an axial flow, two stage turbine which employs superheated steam in series with the main power plant turbo generators.

One concept is similar to the HTGR circulator in which the circulator and turbine are on a common shaft in an integral unit which contains a water-lubricated bearing system that is isolated from the main helium circuit by an auxiliary flow of purified buffer helium through a labyrinth seal at the compressor end of the shaft. A buffer water system at the turbine end of the shaft prevents high pressure steam from entering the bearing water system.

A second concept consists of a circulator unit similar to the integral unit but with the turbine removed. The turbine drive is in a separate external drive unit utilizing a conventional oil lubricated bearing system. This concept was developed to simplify circulator test facility and reactor hot flow tests so that the circulator can be driven by an electric motor drive for these tests without the need for large auxiliary boilers and condensers or a steam compressor system. Both of the above concepts are applicable to both vertical and horizontal installations in the PCRV.

As part of the circulator design effort, an optimization study was made of steam turbine blades to obtain high efficiency and low pressure loss in the turbine, as well as practical dimensions for ease of manufacturing. Similarly, an optimization study was made of the compressor blade design to obtain high efficiency and blade speeds which would yield acceptable blade dimensions and speeds on the drive turbine.

The two stage circulator and turbine drive both have more acceptable stress levels than the single stage design used in previous reference designs. Also, most critical design parameters on the blades have been changed to more conservative levels in the two stage design. The bearings, shaft and bearing housing are basically the same for both the single and the two stage machines. However, the rotating speed is reduced from 8400 RPM for the single stage to 6000 RPM for the two stage design. Overall, each stage of the two stage circulator and turbine drives is within the state-of-the-art of technology for turbo machinery.

2. DESIGN AND OPERATING REQUIREMENTS FOR MAIN CIRCULATOR SYSTEM

The function of the main helium circulators is to generate the pressure rise necessary to circulate the required flow rates of primary coolant through the core and the primary loops.

The circulators are required to operate continuously in the dry helium primary coolant atmosphere for a design service life of 30 years. The circulators must be capable of operating under various normal and abnormal conditions, including:

1. Normal plant operation between full load and minimum load.
2. Plant start-up.
3. Routine plant shutdown for refueling or maintenance.
4. Anticipated operational transients.
5. Steam leak or air ingress following accidental depressurization.

The circulator operating parameters for pressurized conditions under 100% load are given in Table 1 and are based on inlet steam conditions upstream of the circulator control valves of 2400 psi and 950°F (16.53 MPa and 515°C). The circulators must be capable of operating without surge instability or surge at all operating conditions. The point of surge, which is defined as an abrupt drop in blower pressure rise at low flows, must occur at a flow less than 70% of the full-load flow at design speed.

Helium coolant flow through each of the three primary loops is accomplished by an axial flow, two stage helium circulator driven by an axial flow, two stage steam turbine. The circulator utilizes a water-lubricated bearing system for the rotor shaft and a helium buffer seal system. The steam turbine for driving the main helium circulator operates in series with the plant main steam turbine. In this flow arrangement, the steam-driven helium circulator tends to be inherently self-regulating since the helium cooling requirements of the reactor are directly related to the supply of steam produced. The principal characteristics of the main helium circulator and its steam turbine drive are given in Table 1. A general arrangement of the integral design main helium circulator is shown in Figure 1. Each unit can be mounted either vertically or

TABLE 1

PRELIMINARY CONCEPT FOR 2 STAGE CIRCULATOR WITH ΔP OF 60 PSI

	CIRCULATOR	TURBINE
OPERATING PARAMETERS		
Inlet Pressure - MPa (psi)	8.61 (1,250)	16.4 (2,380)
Inlet Temperature - °C (°F)	301 (577)	512 (943)
Outlet Pressure - MPa (psi)	9.03 (1,310)	9.13 (1,325)
Outlet Temperature - °C (°F)	316 (600)	419 (785)
Flowrate - kg/sec (lbs/sec)	247 (542)	127 (278)
Horsepower - MW	17.4 (23,270)	17.4 (23,700)
Efficiency	80	82
Tip Speed - m/sec (ft/sec)	297 (974)	200 (658)
BLADE CHARACTERISTICS		
Tip Diameter - mm (in.)	945 (37.20)	636 (25.06)
Hub Diameter - mm (in.)	802 (32.20)	609 (24.00)
Blade Height - mm (in.)	63 (2.50)	14 (0.53)
Blade Width - mm (in.)	42 (1.63)	25 (1.0)
Aspect Ratio	1.25/1.25	.53/.50
Solidity	2.0	1.6
Up. Str. Vel. - m/sec (ft/sec)	198 (650)	92 (300)
Down Str. Vel. - m/sec (ft/sec)	---	136 (446)
Mean Diffusion/Reaction	.38	.04
Mean Camber	.69/.69	5.2/3.6
Nozzle Mach No.	---	.58/.65
DISK SHAFT & BEARING DIMENSIONS		
Rim Width Tip - mm (in.)	47 (1.85)	31 (1.2)
Disk Width, Ctr. mm (in.)	51 (2.0)	25 (1.0)
Shaft Diameter - mm (in.)	203 (8.0)	153 (6.0)
Journal Bearing Diameter - mm (in.)	229 (9.0)	229 (9.0)

Table 1 (continued)

	CIRCULATOR	TURBINE
STRESS LEVELS		
Blade Cent. Stress - MPa (psi)	46.1 (6,674)	13.0 (1,878)
Blade Bend. Stress - MPa (psi)	64.1 (9,300)	37.2 (5,400)
Blade Root Stress - MPa (psi)	91.0 (13,200)	75.8 (11,000)
Rim Lobe Stress - MPa (psi)	135.0 (19,600)	62.8 (9,100)
Disk Stress - MPa (psi)	179.0 (26,000)	82.8 (12,000)
Allowable Tensile Stress - MPa (psi)	275.5 (40,000)	275.5 (40,000)
SPEED CONDITIONS		
Blade Passing Freq. - CPS	400	12,800
Blade Fundamental Resonance - CPS	1,500	18,000
Shaft Critical Speed - 1%	146	146
Shaft Speed - RPM	6,000	6,000
Allowable Overspeed - %	124	124
Momentary Overspeed - %	135	135

horizontally in the prestressed concrete reactor vessel (PCRV) primary closure.

Helium enters the compressor through an accelerating inlet section and discharges into a center duct and is decelerated in a diffuser. The helium flows into the upper plenum above the core from the circulator outlet duct.

Steam enters from the upper penetration piping and flows downward through the center pipe of two concentric pipes until entering the nozzle inlet. Next, it expands through the nozzle ring and turns in the turbine blades to provide power to drive the helium compressor. The steam discharges from the turbine blades into a radial annulus turning 180° prior to entering the diffuser where it is decelerated and discharged to the balance of plant system through the outside pipe of the two concentric pipes leading into and out of the circulator turbine.

The circulator and the steam turbine disks are mounted on a single shaft overhung from opposite ends of a central housing that contains the thrust and journal bearings and shaft seals. The bearings are of the water-lubricated type. A schematic illustration of the internal arrangement of the circulator is shown in Figure 2.

The circulator radial and thrust bearings are of the shrouded step type. This type of bearing was selected because it provides a large hydro-dynamic lift and is suitable for operating hydrostatically without undue complication. A typical bearing pad and the resulting pressure profile are shown in Figure 3. During operation, water under pressure is supplied to the pocket via the inlet feed groove to support the bearing load. Rotation generates pressure in the pocket before the step where the maximum pressure results. The purpose of the lands is to limit side leakage from the pocket, and this shrouding action gives the bearing its name. The bearing operates in both the hydrostatic and hydrodynamic regimes and is therefore a hybrid. The tangential flow of water causes rotation of the rotor when the bearing water is turned on. A brake is used to keep the rotor from turning when the

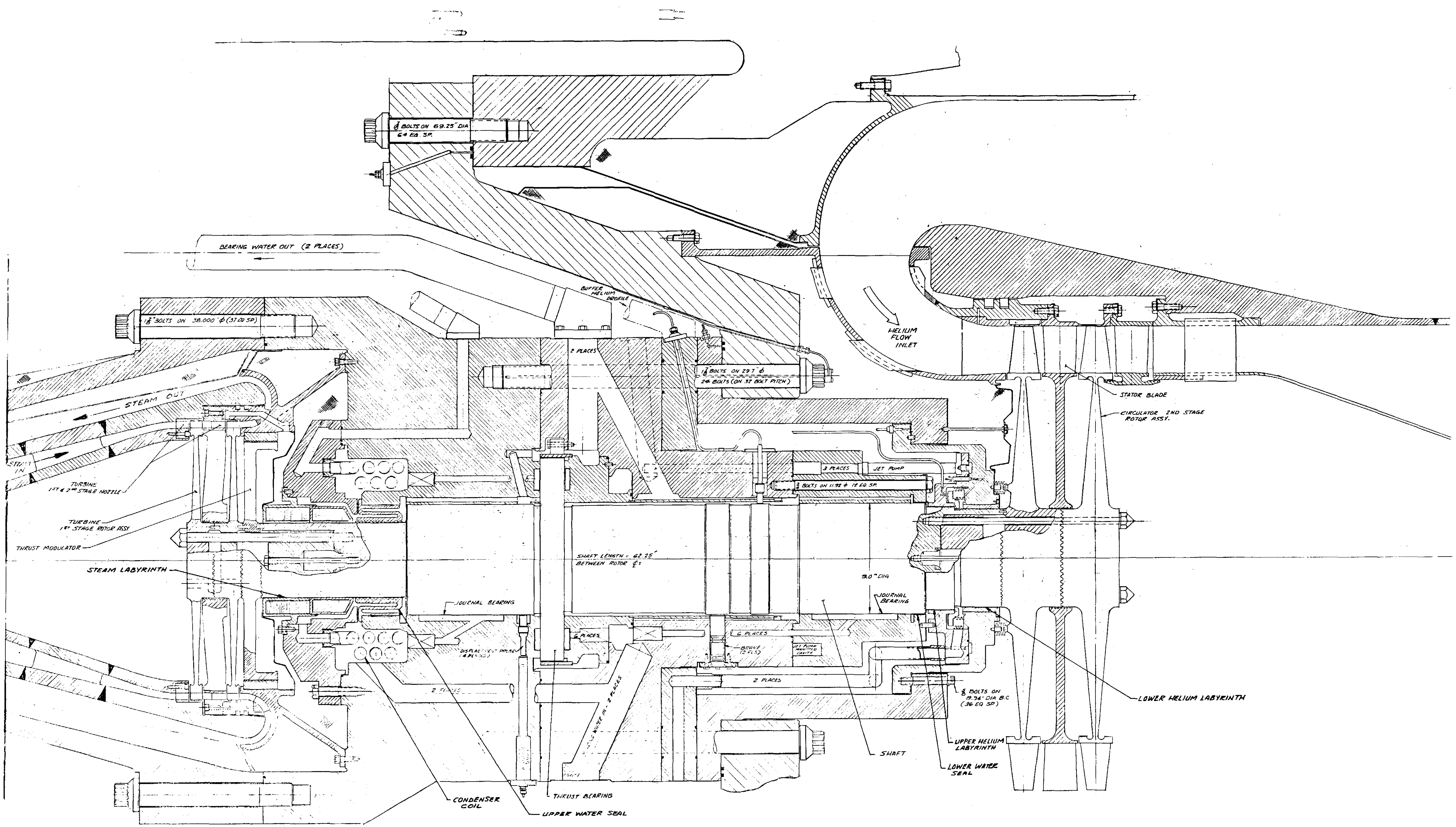


Fig. 1. Main circulator layout

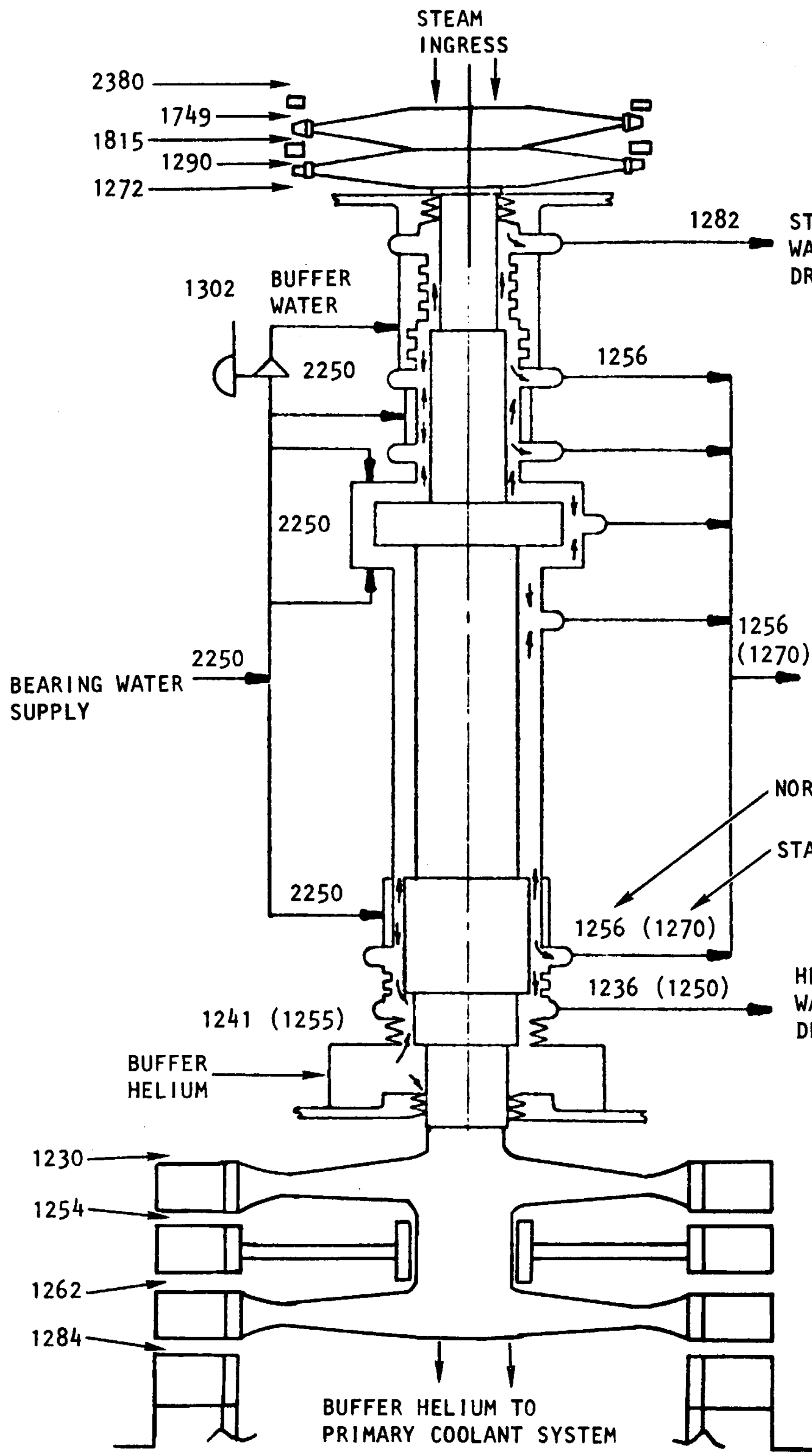


Fig. 2. Circulator bearing system pressures (psi or 1/6900 Pa)

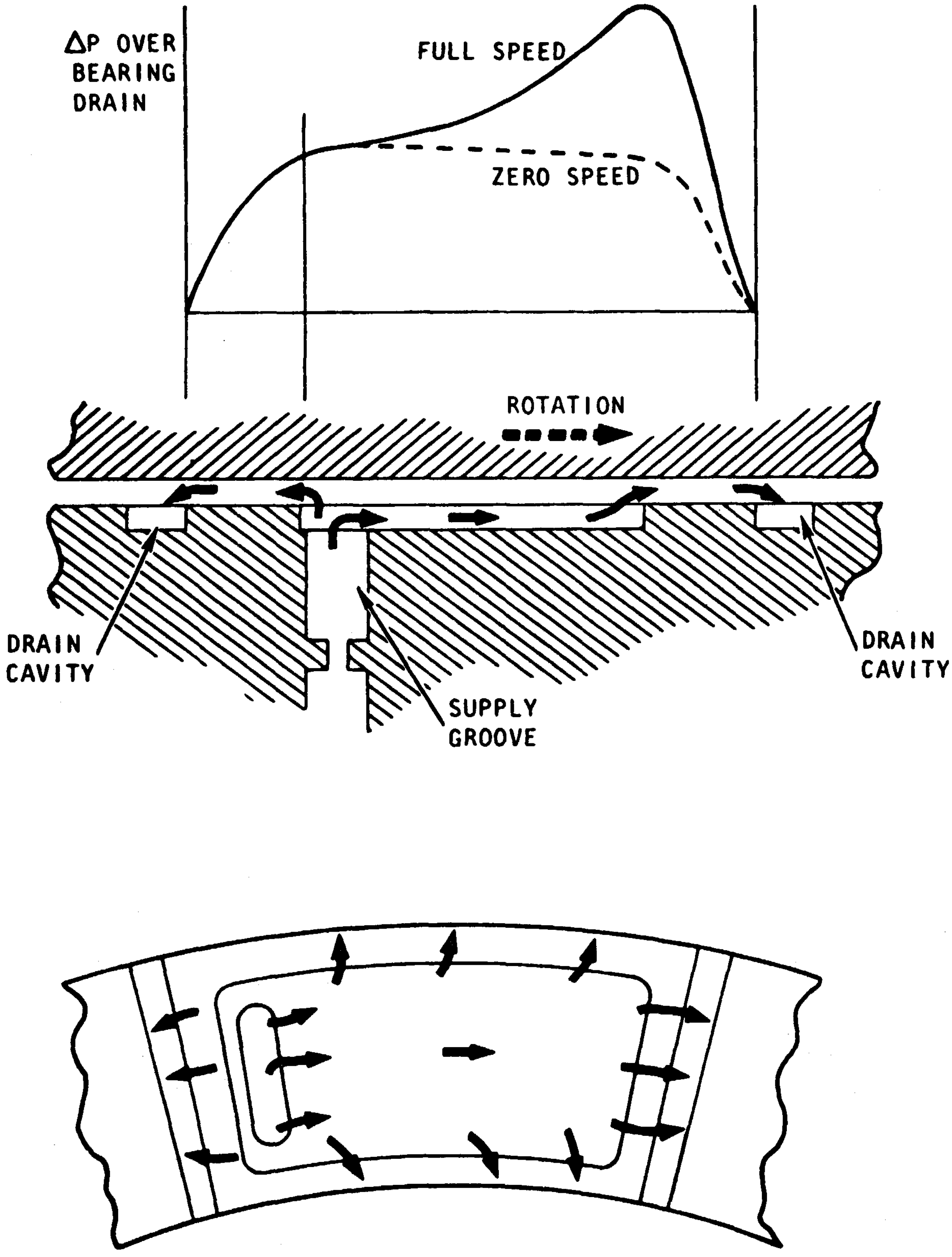


Fig. 3. Shrouded-step bearing pad

operator desires it. The journal bearings are similar to the thrust bearing, with their pads curved along a cylindrical surface.

A double-labyrinth seal is provided between the compressor disk and the compressor-side journal bearing. A labyrinth seal is located between the turbine-side journal bearing and the steam system and is arranged so that leakages are toward the steam-water scavenge chamber and drain system which transfers the mixture to an external accessory system after recycling.

The circulator is controlled by means of a steam-turbine throttle valve located outside the PCRV and above the circulator in the high-pressure superheated steam line from the steam generator in the loop in which the circulator is located.

The circulator performance characteristics are presented in Figures 4 and 5. These figures show the pressure rise ratio and normalized horsepower requirements versus normalized helium flow rate for various shaft speeds. The design point is 6000 rpm with a helium flow of 542 lb/sec (247 kg/sec) and a pressure rise ratio of 1.048, requiring 23,270 hp (17.4 MWatts).

The turbine performance characteristics are presented in Figure 6, which shows the normalized turbine enthalpy drop and pressure drop ratio versus normalized steam flow rate for various shaft speeds. The design point is 6000 rpm with a steam flow of 278 lbs/sec (127 Kg/sec) and an enthalpy drop of 60.35 Btu/lb (141 KJoule/kg).

A two stage circulator driven by a two stage drive turbine was chosen for this design instead of the single stage configuration used in the past. Both a two stage circulator and a two stage drive turbine require lower rotational speeds with the tip diameters and blade heights approximately the same. Also, most other parameter changes are insignificant. It is therefore advantageous to have the same number of stages on both the circulator and the drive turbine. If only the circulator were changed to two stages then the rotational speed would have to be reduced and the turbine diameter increased with the result that the turbine blades would be unacceptably short. Likewise, a two

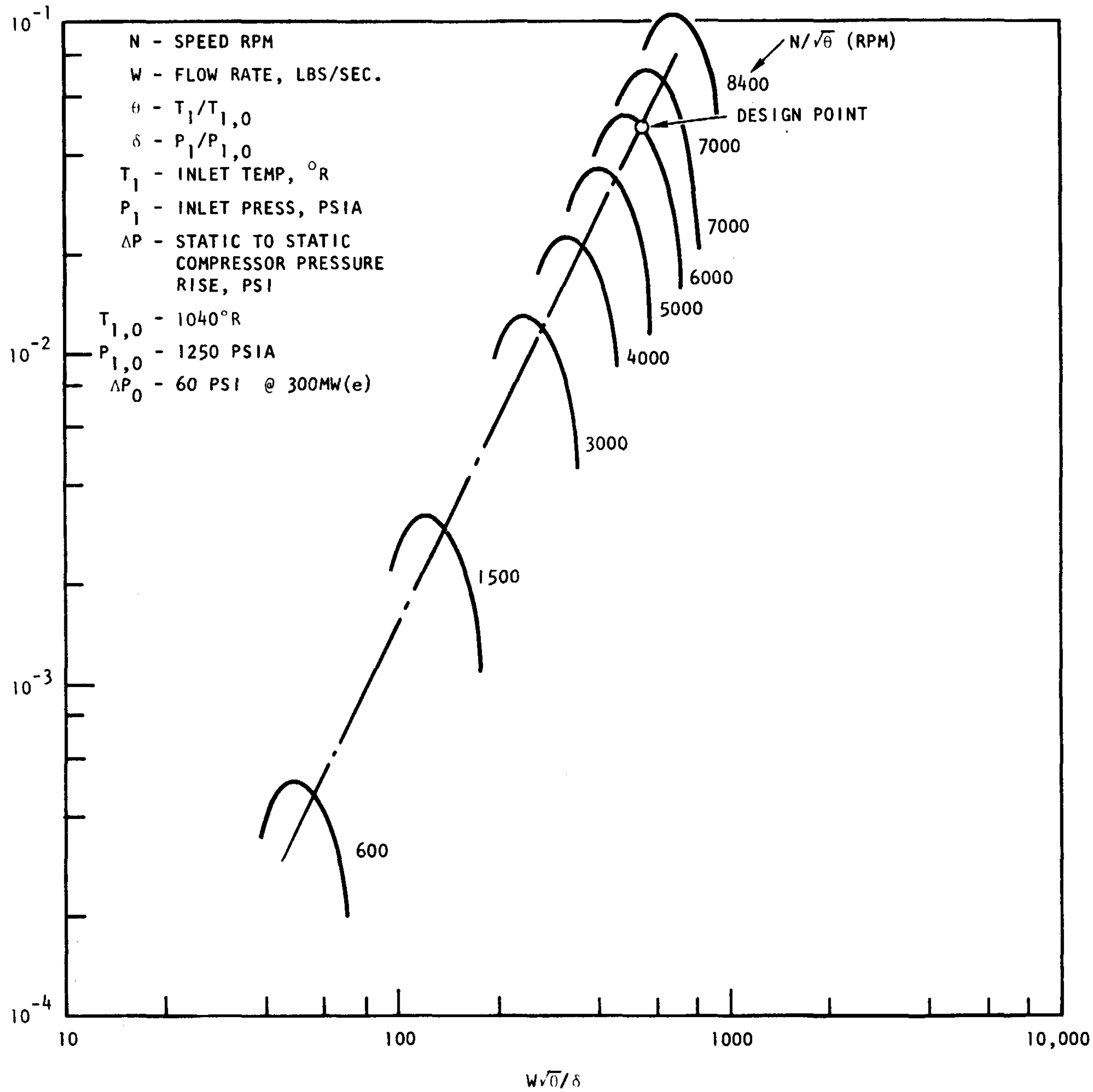


Fig. 4. Pressure rise ratio versus normalized helium flow rate

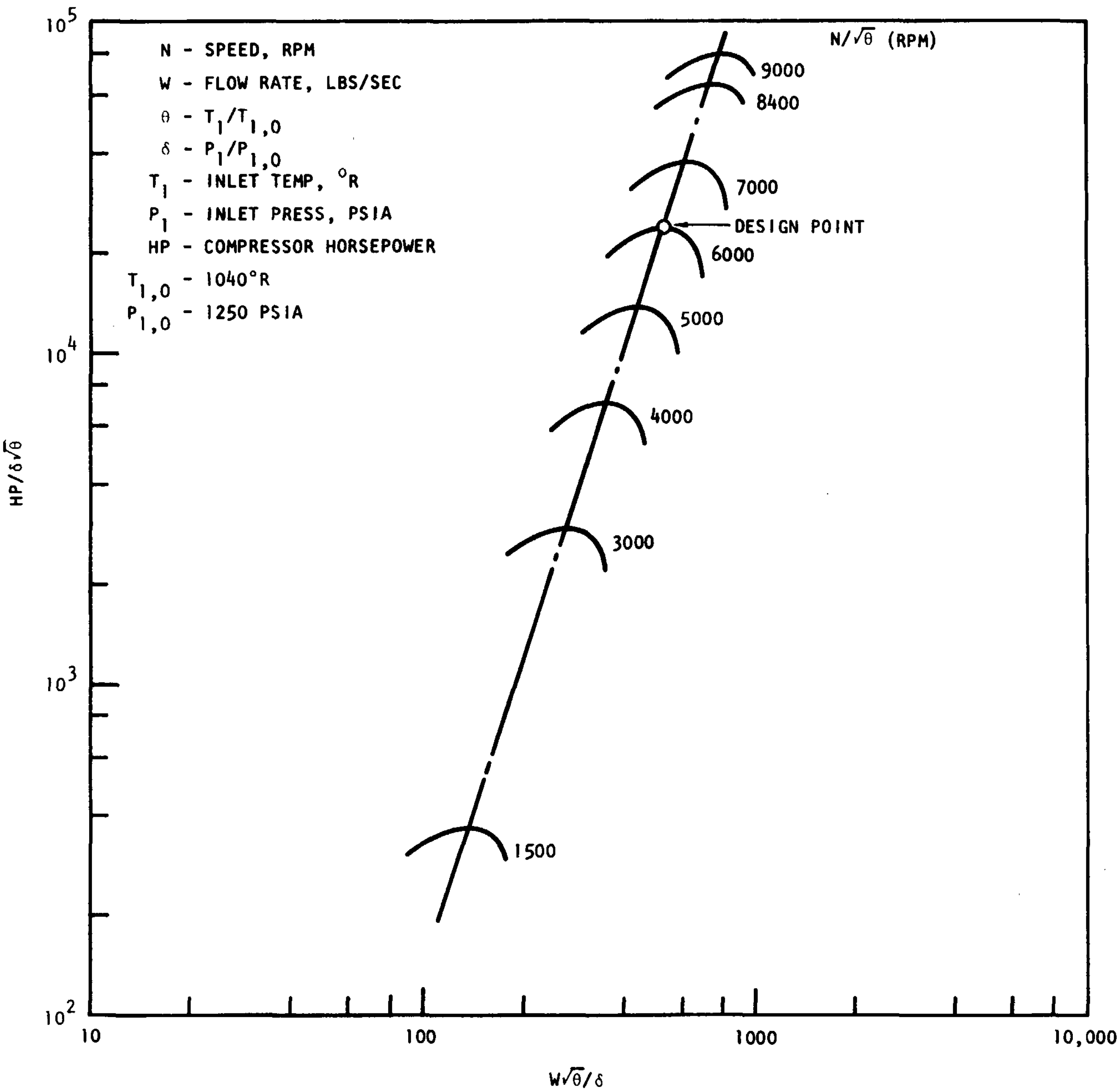


Fig. 5. Normalized horsepower versus normalized helium flow rate

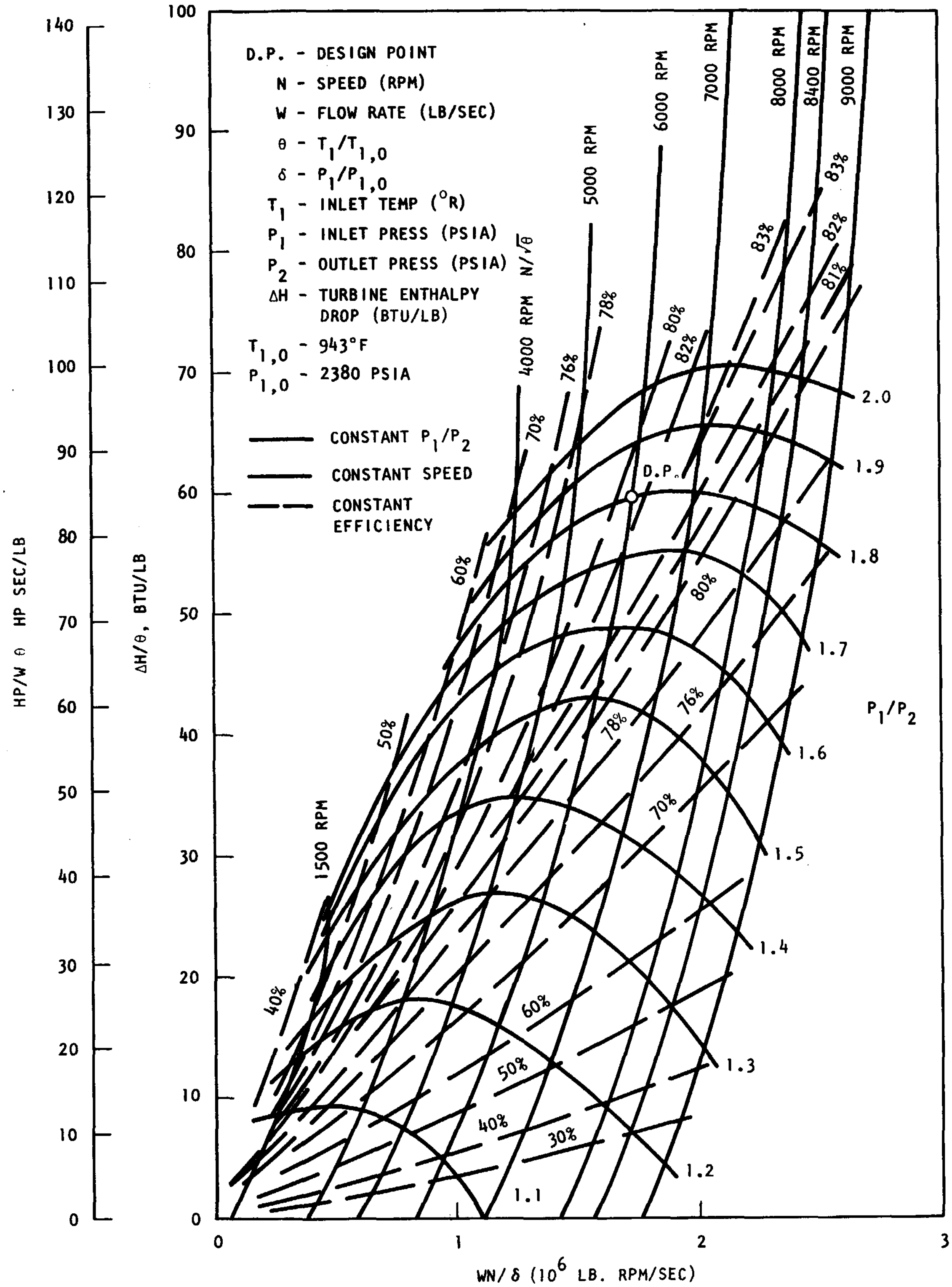


Fig. 6. Normalized drive turbine data

stage turbine would require an undesirable increase in circulator diameter and shorter blades. The analysis was therefore limited to investigate a two stage circulator driven by a two stage turbine. In order to obtain minimum thrust loads on the shaft it is necessary to have a pure impulse turbine which results in a reduction of the circulator speed from 8,400 rpm to 6,000 rpm.

The two stage circulator and turbine drive both have much lower stress levels than the single stage design. Also, most critical design parameters on the blades have been changed to more conservative levels in the two stage design. The bearings, shaft and bearing housing is basically the same for both the single and two stage machines. Overall, each stage of the two stage circulator is within the technology established for the HTGR circulators. Table 2 summarizes the improvements obtained with the two stage circulator design. Table 3 describes alternate two stage circulator design conditions.

TABLE 2

COMPARISON OF TWO-STAGE VS. SINGLE-STAGE CIRCULATOR*

($P = 60$ psi, P_{He} - 1250 psi, P_{steam} - 2380 psi)
(.41 MPa) (8.61 MPa) (16.4 MPa)

CIRCULATOR

- Stress levels in blades reduced by factor of two and are no longer critical
- Stress levels in disk reduced by 1.75
- Shaft critical speed increased to 146% of design speed (vs. 138%)
- Allowable continuous overspeed 124% (vs. 110%)
- Improved diffusion factor .38 vs. .51
- Design of each stage similar to HTGR
- Improved natural blade frequencies vs. imposed frequencies

TURBINE

- Stator Mach numbers reduced to .58/.65 as compared to .93 previously
- Stress reductions and frequencies similar to circulator improvements
- Overall thrust loads are reasonable
- Bearings/shaft dimensions unchanged
- Bearing performance now within HTGR and computer code limitations

*As compared to single stage design documented in GA-D12974, "Preliminary Design of a Helium Circulator and Steam Turbine for a 300 MW(e) GCFR Demonstration Plant"

TABLE 3

ALTERNATE TWO STAGE CIRCULATOR DESIGN CONDITIONS

To obtain a more compact circulator:

- Reduce bearing/shaft diameter
- Reduce bearing flow rates and pressures
- Reduce disk weights
- Increase rotating speed
- Increase allowable stress levels to limits
- Reduce allowable overspeed to 110%

To obtain high allowable overspeed:

- Increase disk stiffness
- Increase shaft/bearing diameters
- Increase bearing flow and pressure
- Critical speeds will be limiting

3. EVOLUTION OF GCFR CIRCULATOR DESIGN

The question has been raised as to the need to design, develop and test seals, bearings, etc., specially for the GCFR circulator rather than utilizing or adapting HTGR components or alternatively procuring existing off-the-shelf components. The question is a valid one, particularly in view of the desirability of taking advantage of the reliability and operating experience of proven components, and the objective of minimizing development costs. The possibility of using prior developed and available equipment for the circulation system needed either by the HTGR or the GCFR was precluded by the specialized nature of the task. This also applied to equipment reliability requirements to meet the 30-40 year service life with minimum maintenance. Special design factors included the high sonic speed and low density of the helium gas, requiring high rotational speed to achieve the necessary pressure rise and the high total drive powers required. Need for speed variation was also a significant factor effecting drive choice.

In the case of the HTGR, both axial and centrifugal circulators were considered, and possible drive systems included both steam turbine and electric drives. The HTGR system selected, single stage axial circulators driven by single stage steam turbines, was most strongly influenced by the resulting very compact and efficient machine form, readily incorporated in the PCRV.

The higher pressure rise and pumping power requirements of the much higher power density cores of GCFR's not only further restricted alternate possibilities but also enhanced other advantages of the integrated turbo-circulator system. Among these were the latters independence from external power sources (reheat steam from the turbine) and ability to continue duty adequately even for depressurized conditions with only the reactor after-heat as an energy source.

This circulator system was particularly desirable for use with the very low heat capacity core of the fast reactor, for, in effect, it utilizes the high heat capacity of the boiler system as an alternate aid to sustain circulation, following even complete external system isolation.

It is for these reasons, and the fact that the technology for circulator and drive systems have now been developed for Fort St. Vrain and the large HTGR, that the basic circulator configurations have been carried forward for the GCFR.

The present GCFR circulator arrangement in the steam cycle was selected on the basis of developing a system that would

(a) satisfy the stringent safety requirements for the breeder reactor,

(b) be economically competitive in terms of cost and plant efficiency,

(c) be reliable,

(d) be based upon technology developed for the HTGR.

An important objective was to develop a system which could guarantee uninterrupted availability of primary coolant circulation following any plant transient condition; thereby, providing the necessary heat removal capacity until the auxiliary cooling system is operable.

This rationale led to the fundamental concept that the circulators should be closely coupled to the steam generators and also be capable of operating independently of the main turbine in the event of a turbine trip.

This situation is contrary to the large HTGR and Fort St. Vrain systems in which the circulator is located in the cold leg of the gas reheat line from the main turbine. In an HTGR system the steam supply to the circulator can be interrupted with subsequent loss of primary coolant circulation through the core. That is possible because the large heat capacity of the graphite core allows the core to be without cooling for some time and not incurring any safety hazard.

In the case of Fort St. Vrain, after steam supply to the circulator is cut off, circulation is re-established by providing water to drive the Pelton wheel mounted on the same shaft as the steam turbine.

On the large HTGR if the main circulators are shutdown within a short time (approx. 5 minutes) independent electrically-powered circulators are started to provide core cooling.

For the GCFR, with its smaller, relatively low-heat-capacity metal clad core, uninterrupted cooling is essential. It is for this reason, together with the high pumping power required, that the circulator is installed in the high pressure steam line immediately downstream from the steam generator. This arrangement plus the available steam conditions also allows the use of a single-stage turbine and compressor to deliver the necessary power and helium flow. However, two stages were chosen for the present reference design to obtain conservative stress levels and blade performance.

In the GCFR reference design, circulator turbine inlet steam is supplied from the steam generator superheater. Circulator turbine outlet steam is then piped directly to drive the main turbine.

The basis for selecting the present axial-flow two-stage turbine and compressor configuration was simplicity, compactness, ability to satisfy all operating requirements and to be within the state-of-the-art for compressor and turbine design. Compactness is important in reducing the penetration size in the pressure vessel and in simplifying removal or replacement.

Additionally, the Fort St. Vrain circulator has been designed, built and qualification-tested with virtually no major changes to the design. More recently a higher power circulator has been designed and built for the larger HTGR reactors which incorporates a number of design improvements and is presently awaiting testing.

3.1 Compressor Design

The aerodynamic configuration of the GCFR main circulator was chosen using basic principles of turbomachinery design. The axial flow compressor is appropriate for low head, high flow applications such as the GCFR primary coolant flow conditions.

The circulator for the GCFR reference design operates extremely close to the best efficiency that can be obtained with any turbocompressor aerodynamic configuration and the axial flow arrangement is compatible with the many other structural and functional requirements of the main circulator.

The pressure ratio $(\frac{\Delta p+p}{p})$ for the GCFR is 1.048 as compared to 1.028 for the HTGR compressor design. A two stage circulator was chosen for the GCFR circulator to maintain conservative operating parameters similar to HTGR. The compressor is designed to satisfy DBDA conditions without surge. In turn, high surge margins are obtained during normal operation. Since flow conditions are not anticipated to change during the reactor life and since adequate margins are included in the designs, it seems reasonable to assume that the GCFR circulator will operate satisfactorily during its design life.

3.2 Turbine Drive

The GCFR two-stage circulator turbine drive is within the state-of-the-art of steam turbine design. The two stages are nearly pure impulse turbines with very low Δ p across each disc. In the current circulator design the turbine steam exit pressure nearly balances the helium pressure acting on the shaft cross sectional area and in the steam and bearing water drains. A multi-stage turbine in series with the main turbine would require a reaction type turbine which has a higher Δ p across the turbine discs and consequently larger thrust loads.

A parallel flow system, in which only a fraction of the steam flow passes through the circulator turbine, would require a larger enthalpy drop and much lower steam exit pressures in order to deliver the same horsepower to the circulator. The lower steam exit pressure would create serious shaft thrust loads due to the much higher helium pressure acting on the shaft cross sectional area. Bearing drain water pressures in this case would also be much higher than the steam pressure creating the potential for large water leakage into the steam system. A multi-stage turbine in a parallel flow system would also have extremely short blades (less than 0.25 inches on the first stage) which would present problems of parasitic losses due to the required clearances and would require 7 to 11 stages. Any multi-stage turbine would be more complex, have larger shaft diameter to compensate for the larger disc weights and overhangs and larger thrust loads due to the helium and steam pressures acting on a larger shaft cross sectional area.

The possibility of using a radial inflow turbine drive for the GCFR main helium circulator with a radial flow compressor was investigated. Only single stage turbines were considered because of the difficulties of routing the steam flow in multi-stage radial flow designs. The radial inflow turbine does not appear suitable for the main helium circulator because:

1) For optimum performance, the turbine tip diameter should be about 3 feet. This results in an unrealistically small tip width and a large diameter steam casing. In addition, with radial inflow, the high pressure inlet steam must be carried in the water/steam drain pipe.

   This combination would require high steam piping wall thicknesses for the radial flow turbine.

2) The radial inflow turbine at 6000 rpm (matching compressor speed) has a low specific speed of about 21, which will result in a low efficiency of about 50% based on normalized performance curves.

3) Because the radial turbine is of the reaction type, high steam pressures must be balanced against the bearing water pressures; therefore, requiring a buffer water system.

3.3 Bearings and Seals

Differences in circulator steam pressure, bearing water pressure and buffer helium pressure between the HTGR and the GCFR circulators are not the result of an arbitrary selection, but are imposed by applying the same design philosophy to the two reactor systems. A diagram showing the bearing and seal pressures in the circulator system is given in Figure 3.

The buffer helium conditions are related to the primary coolant system conditions. By definition the buffer helium must be at a higher pressure in order to provide an inflow of clean helium into the primary system, and to prevent the leakage of contaminated (active) helium into the bearing water/helium drain system. The bearing water pressure is determined by the stiffness requirement of the shaft journal to obtain a first order critical speed approximately 40% higher than the operating design speed (increased shaft stiffness is insufficient). In addition, the bearing water pressure at the drain end of the journal bearing must be close to that of the buffer helium pressure at the upper helium seal to prevent excess leakage of water or helium.

For these reasons, the bearing water pressure for the GCFR is approximately 2,250 psi (that is 1,000 psi above the primary coolant system pressure of 1,250 psi) compared to a bearing water pressure of approximately 1,450 psi for HTGR (i.e., 750 psi above the primary coolant pressure of 700 psi).

At the turbine steam end of the circulator, a similar relationship is required between the steam pressure, at the steam drain end of the steam seal, and water outlet pressure from the upper journal bearing.

In the case of the HTGR the circulator steam is cold reheat steam from the main turbine at approximately 700°F (376°C) and 900 psi (6.2 MPa). After the pressure drop through the steam seal, the steam is at approximately the same pressure as the water at the drain end of the upper journal bearing. Similarly in the GCFR, after the high pressure steam from the superheater (950°F and 2400 psi) (515°C and 16.53 MPa) has dropped through the steam seal it is at approximately the same pressure as the bearing water at the steam/water drain.

Another very important relationship affecting the difference between the steam and the primary helium pressure is the desirability to ensure that axial thrust loads at each end of the rotor shaft are reasonably balanced. This is required to avoid excessive thrust bearing loads in the circulator. Again this balance is related to the bearing water pressure through the net thrust loads and the thrust bearing size.

From the foregoing it can be seen that the steam, water and buffer helium with the respective primary coolant system pressures are compatible for both the GCFR and HTGR. However, GCFR primary system pressure is twice that of HTGR; therefore, if similar circulator system conditions were imposed the GCFR primary pressure and steam pressure would have to be degraded to the HTGR levels. This of course is not practical and it is therefore not possible to use HTGR bearings directly in the GCFR circulator.

3.4 Steam Cycles

Steam cycle studies have been made for the large HTGR to ascertain the technical, efficiency, and economic aspect of the alternate systems. Steam system studies have included configurations with the circulator installed in the cold gas reheat line (per Fort St. Vrain), parallel gas reheat system (low pressure circulator turbine); parallel gas reheat (high pressure circulator turbine); non-reheat system (serves high pressure circulator turbine); and, non-reheat (parallel flow circulator turbine).

These studies have covered a wide range of plant sizes and circulator steam conditions and have heavily concentrated on the operating efficiency and plant capital cost aspects. Such studies and the associated trade-offs are also highly specialized since they relate to the primary coolant pressure steam cycle temperature and pressure and the related vessel and equipment costs and cannot be related to GCFR conditions in a direct way. It is interesting however, to note that the optimum trade-off between these various systems for the HTGR helium and steam conditions appear to favor the gas reheat cycle with the circulator in the cold leg of the reheat circuit from the HP turbine to the reheater.

The reason for this conclusion is not circulator related but steam generator related. With the high gas temperature encountered in the HTGR (1400°F, 760°C range) it is not acceptable to pass this hot gas directly over the superheat tubes. Due to the high steam pressure (2500 psi or 15 MPa) tube walls are extremely thick (approximately 0.25 inches for a 1" diameter tube, 6mm for a 25 mm diameter tube) and resulting thermal stresses are unacceptable. Therefore, gas reheat is employed in the hot gas region which has lower steam pressure and resulting lower tube wall thicknesses. With the reheat steam readily available in the HTGR it is economical to also drive the circulators with this steam.

In the case of GCFR, the higher horsepower requirements, requirements for uninterrupted cooling and high plant efficiency favor the series flow steam arrangement in which all of the steam from the steam generator superheater flows through the circulator drive turbine before entering the plant main turbine. Helium temperatures and steam pressures and temperatures are acceptable for this series flow arrangement in the GCFR.

3.5 Circulator with External Drive

A concept using an external turbine drive unit was also investigated, Figure 7. The circulator unit would be identical to the integrated circulator design in Figure 1 except that the two turbine stages are missing, Figure 8. The external turbine drive unit, Figure 7, contains two turbine stages exactly the same as the ones used in the internal design in Figure 1. Oil lubricated bearings are used in the external unit for simplicity of the service system for this unit.

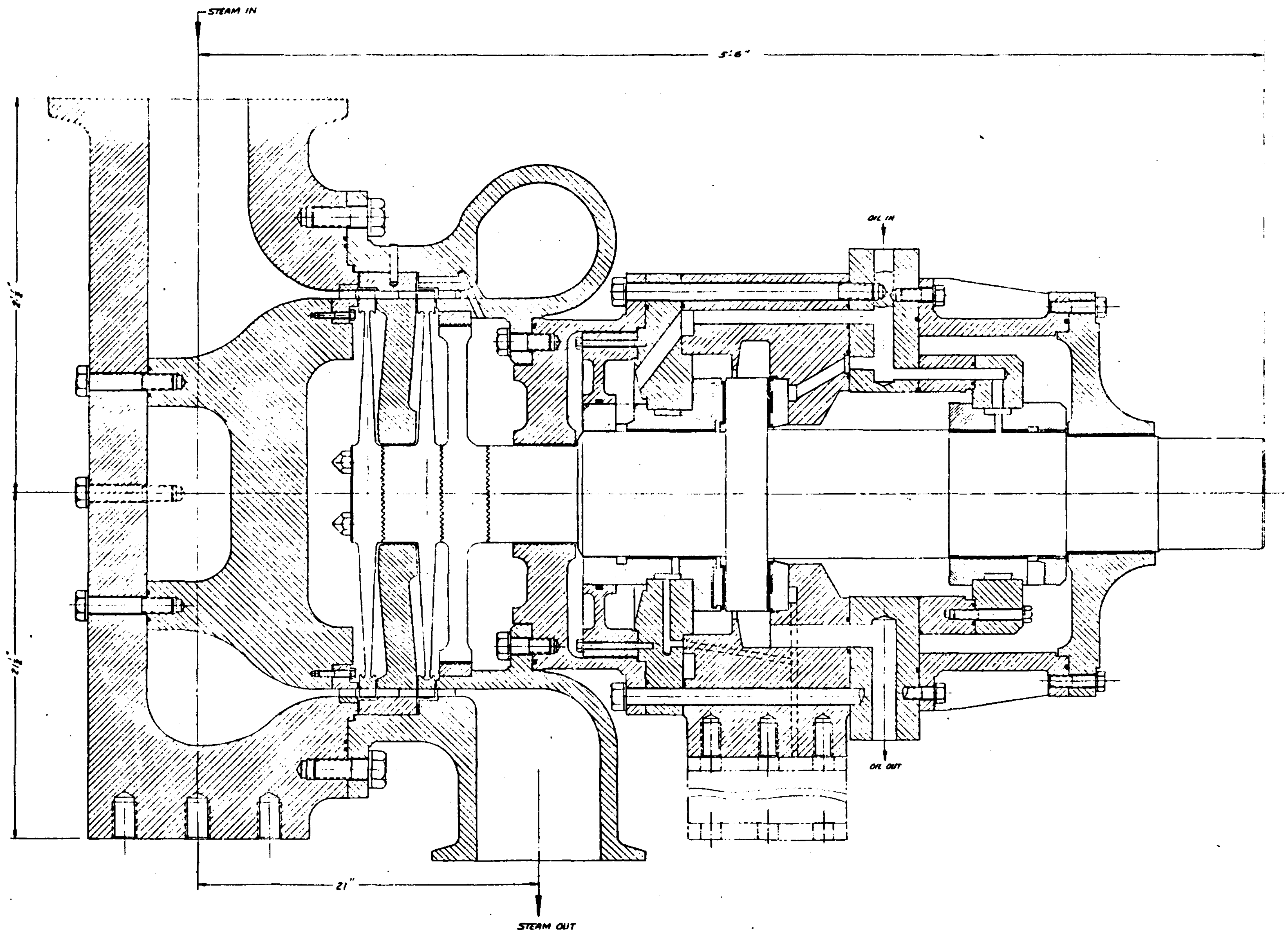


Fig. 7. External series flow drive turbine unit

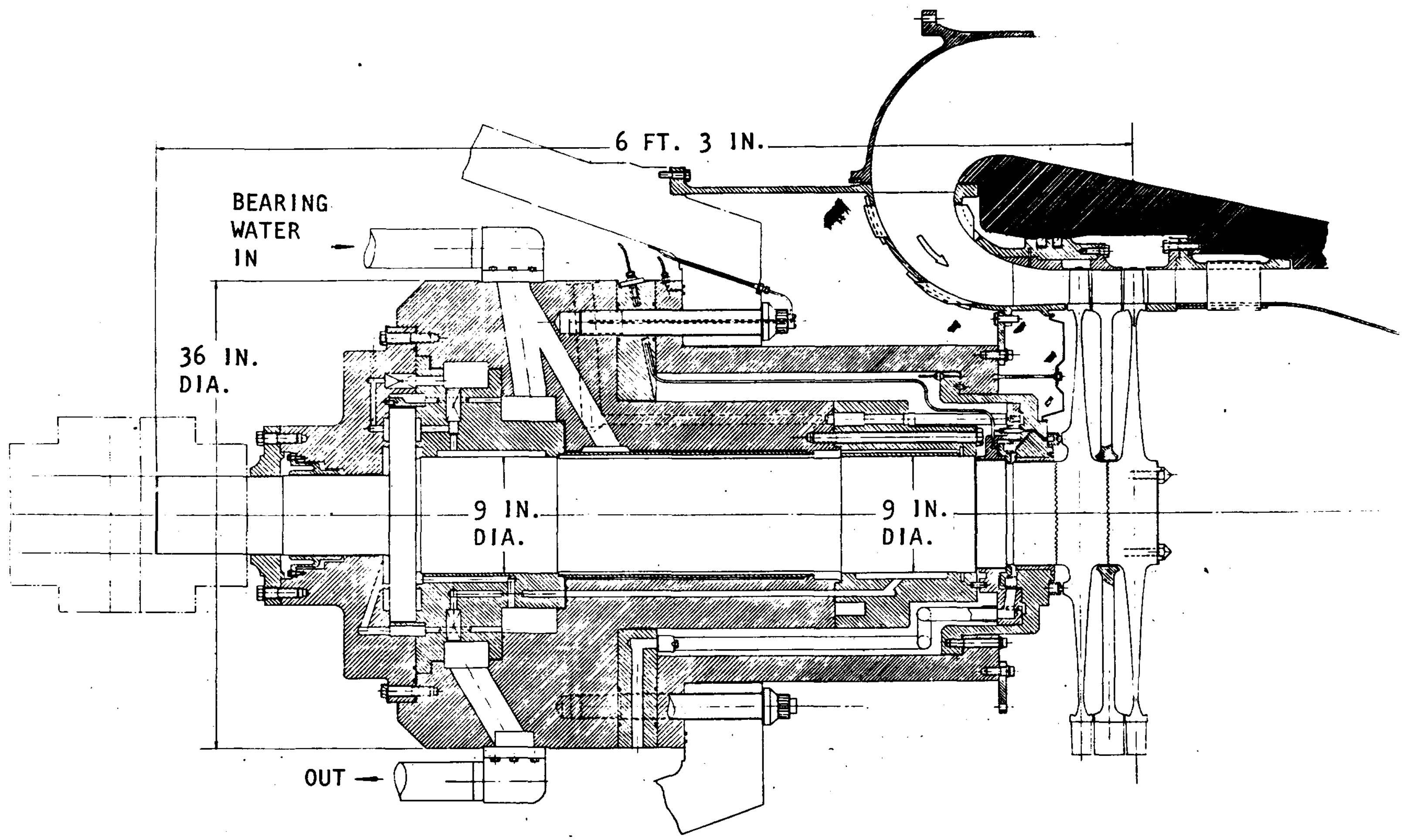


Fig. 8. Axial compressor with water lubricated bearings and using external steam driver

This arrangement has a number of advantages. (1) Any future variations in performance requirements will have minimal effect on the overall concept, (2) it satisfies both demonstration plant and commercial plant requirements, (3) both compressor and turbine are well within the state-of-the-art of conventional technology, (4) the performance predictions are more conservative, (5) the external drive arrangement is very compact as compared to electric drive and multi-stage parallel flow turbines, (6) it fits both in a vertical and in a horizontal installation, (7) only the compressor needs to be tested in a test facility since the turbine can be tested separately by a turbine manufacturer. The test facility can therefore be simplified by using a direct electric motor drive or gas turbine drive. Hot flow tests in the reactor can be done in the same fashion, (8) high cycle efficiency, compactness and the safety of using residual heat for steam generation during emergency shutdown can be maintained without outside power or emergency power and with no interference with main plant turbine operation.

The disadvantage is mainly related to having two separate units for the circulator and drive turbine. Each unit is almost as big as the integral circulator unit. However, the separate drive turbine unit is much smaller and simpler in design than electric drive motors or multi-stage drive turbines.

3.6 Circulator Similarity for Commercial Plant

The circulator for a commercial plant will be similar to the demonstration plant circulator except larger in size to provide 2 to 2.5 times the horsepower. The overall diameter will be only slightly larger with longer blades and lower rotational speed. The commercial plant circulator will follow exactly the same development plan as for the demonstration plant. The only difference will be the need for a test facility which can accommodate the larger helium flow rates and steam flow rates to obtain the large horsepower. Steam and helium pressures and temperatures will be identical. If the test facility is powered by power plant steam, there are a number of power plants, Table 4, with unit boiler capacities larger than required for the commercial plant circulators. If a steam compressor system is used, two steam compressor units of the type used for the demo-plant circulators will be required. However, if an external drive system is used for the circulators for both demo and commercial plants, a number of simpler alternative drive systems are possible for the test facilities.

TABLE 4

TYPICAL POWER PLANTS MATCHING CIRCULATOR TEST FACILITY STEAM REQUIREMENTS

TYPE GCFR	NO. CIRC.	PER CIRCULATOR 1000 HP	10^6 LBS/HR	PSI	TYPICAL PLANT LOCATION (NO. UNITS)	CAPACITY PER UNIT 10^6 LBS/HR	PSI	^oF
300 MW(e)	3	20	1	2400	ALAMITOS (1)	1.2	2450	1000
				2400	SO. BAY (3)	1.4	2400	1000
				2400	HUNTINGTON (104)	1.6	2450	1000
1200 MW(e)	6	40	2.0	2400	ALAMITOS (5)	3.3	2450	1000
				2400	ALAMITOS (3)	2.3	2450	1000
				2900	MOJAVE (1)	5.6	3633	1000
1500 MW(e)	6	50	2.5	2400	ALAMITOS (3)	2.3	2450	1000
				2900	MOJAVE (1)	5.6	3600	1000
				2900	REDONDO (7)	3.3	3600	1000
2000 MW(e)	8	50	2.5	2400	ALAMITOS (3)	2.3	2450	1000
				2900	MOJAVE (11)	5.6	3600	1000
				2900	REDONDO (7)	3.3	3600	1000
2000 MW(e)	6	67	3.3	2900	ALAMITOS (3)	3.3	3600	1000
				2900	MOJAVE (1)	5.6	3600	1000
				2900	REDONDO (7)	3.3	3600	1000

4. AERODYNAMIC DESIGN OF MAIN CIRCULATOR SYSTEM

4.1 Matching Turbine-Circulator Requirements

The steam turbine is designed to match the circulator and bearing horsepower requirements at design conditions given in Table 1. During any other operating conditions, the turbine flow and pressure are adjusted to match the circulator and bearing horsepower and speed. However, there are several limitations on the turbine operating conditions, some of which are discussed below. Overspeed and critical speed conditions are discussed in Section 5.3. Table 5 summarizes the parameter restrictions for a circulator and steam turbine drive on a common shaft. The current design is optimized to satisfy the limitations and match the given performance requirements.

4.2 Axial Flow Circulator Design

The main helium circulator for the 300 MW(e) GCFR consists of a two stage, axial flow compressor driven by a variable speed steam turbine using superheated steam in series with the main power plant turbine.

The circulator design is based on using Type 422 stainless steel for the compressor blades. A set of compressor and turbine design parameters were obtained, as shown in Table 1. These design parameters are considered practical and safe for a GCFR circulator. The selection of this set of parameters was based on an optimization analysis of the compressor and turbine variables and on practical design limitations for the manufacture and assembly of the circulator components. A summary of this design study is given below.

The compressor blade roots and disc rim grooves will start yielding at 144% of design speed. Thus, sufficient safety margin is built into the compressor blade. Because of low stress levels at operating speed, it is possible to increase the speed to obtain a

TABLE 5

PARAMETER RESTRICTIONS FOR GCFR MAIN CIRCULATORS

	Compressor	Turbine	Limitations
Speed, N	$f(\sqrt{\Delta p_C}/D_C)$	$f(N_C)$	Fixed by Δp_C and D_C. 1. Decreased speed would increase diameter and result in too-short turbine blades. 2. Increased speed would decrease diameter and result in excessive blade stresses.
Tip diameter, D	$f(\sqrt{\Delta p_C}/N_C)$	$f\ \frac{\sqrt{\Delta p_C}}{N_C}\ \frac{W_T}{h_T}$	Fixed by Δp_C, N, W_T, ΔH. 3. Increased D_C would result in too-short turbine blades. 4. Decreased D_C would result in excessive blade stresses, difficult blade attachments, and use of titanium.
Blade height, h	$f(W_C/D_C)$	$f(W_T/D_T)$	5. Increased D_C would result in too-short turbine blades. 6. Decreased flow rates would result in too-short turbine blades.
Pressure rise, Δp	Δp_C fixed	$\Delta H = f(\Delta p_C W_C)$	7. Δp_C limited by blade loading. 8. Increased Δp_C would require two stage turbine or supersonic turbine.
Flow rate, W	W_C fixed	W_T fixed	Given requirements. 9. See item 6 above.

TABLE 5 (continued)

	Compressor	Turbine	Limitations
Shaft diameter, D_s	$f(L_s, W_d)$	$f(L_s, W_d)$	10. Decreased shaft diameter would cause first critical speed in normal operating range. 11. Increased shaft diameter would cause undesirable axial thrust loads from helium and steam pressures on end of shaft.
Shaft length, L_s	Design	Design	12. Increased length would cause increased shaft diameter and axial loads. See item 11 above.
Blade materials	422 SS	422 SS	13. Changing to higher-strength materials would shorten circulator creep life below 30 yr. 14. Change to titanium would cause lighter blades, slimmer disks, smaller shaft diameter, and smaller axial thrust loads, but also risk of embrittlement in helium atmosphere.

Definitions

D_C = circulator tip diameter

D_s = shaft diameter

D_T = turbine tip diameter

ΔH = turbine enthalpy drop

h_T = turbine blade height

L_s = shaft length

N = circulator rotational speed

Δp_C = circulator pressure rise

W_C = circulator helium flow rate

W_d = weight of disks

W_T = turbine steam flow rate

higher pressure rise during certain depressurized conditions should this become desirable.

4.2.1 Circulator Calculation Method

The computer code AXCOMP, which calculates a complete axial flow circulator blading design, was utilized in a parametric study to establish the compressor configuration. AXCOMP is based on several assumptions including the following:

1. Every segment along the radius of the blade must be capable of imparting equal work to the gas and maintaining a constant axial through-flow velocity. This is known as a free vortex machine.

2. The gas has no radial velocity during its traversal of the blade.

The first step in the design process is to determine the pressure rise required of the compressor. The design pressure rise is then adjusted for diffuser losses. From this modified pressure rise and a guessed polytropic efficiency, the corresponding temperature rise and thus the work required from the blades are determined.

The blade frictional and boundary layer losses are calculated as a function of the diffusion factor, which indicates the extent of boundary layer buildup on the suction surface of the blade.

The AXCOMP code can be used for single stage or multistage machines; either rotor-stator or stator-rotor blading sequences; determination of blade friction and boundary layer losses; alternate blade taper characteristics; and part-load performance. The code output includes the number of blades, blade twist, aerodynamic loading factor, machine efficiency, power requirements, and blade stresses.

4.2.2 Circulator Aerodynamic Design

A parametric optimization study of the compressor section of the circulator showed that it is easier to satisfy the required compressor performance within broader limits of the parameters than for the turbine. The chosen parameters listed below were found to yield optimum circulator performance.

Circulator speed	6000 rpm
Hub diameter	817 mm (32.2 inches)
Blade height	64 mm (2.50 inches)
Axial flow velocity	198 mm/sec (650 ft/sec)
Circulator overall efficiency	80.4%
Blade solidity	2.0

Since rotating speed is not too critical to the circulator performance and blade design the rotating speed was picked to obtain the best turbine performance. The hub diameter was chosen to be big enough to allow a good design of the large bearing housing and sufficient space for the inlet duct and outlet diffuser. On the other hand, to obtain high efficiency and preferable blade dimensions, the hub diameter becomes fixed. Axial flow velocity showed the highest efficiency and the best overall performance at 650 ft/sec in combination with the chosen rotating speed and hub diameter.

From the preceding discussion it is evident that any design choice is necessarily a compromise of many complex, interacting factors. One of the principal factors affecting the GCFR compressor design was the requirement for a relatively large pressure increase across the blading, which in turn influenced blade size and blade bending stresses.

Another limiting factor was a low centrifugal blade stress at the design point to provide adequate circulator overspeed capability for emergency requirements. The design centrifugal blade stress was set at a conservative level to cope with any such credible occurrence.

A third important factor was the desire to produce a compact, economical design that would not create unreasonable bearing demands and thereby inhibit a major objective, reliability.

The final compressor design parameters are listed in Table 1. The compressor performance characteristics are presented in Figures 4 and 5, which show the pressure rise ratio and normalized horsepower requirements versus normalized helium flow rate for various aerodynamic speeds, respectively. The design point for both curves is 6000 rpm with a helium flow of 542 lb/sec, a pressure rise of 1.048 and a required horsepower of 23,270.

4.2.3 Circulator Blade Vibrations

Pertinent data were taken from compressor and turbine computer design calculations. Aerodynamic design considerations such as design point and off design point, number of blades, stators, nozzles and blade profile geometry to meet design criteria are given by the AXCOMP and DSTURB codes. The geometric output, in particular stiffness and mass matrices, of these codes were then used as input for determining the four lowest frequency modes using the SHAKE program for natural frequencies.

The computed natural frequencies were then plotted on interference diagrams (Figures 9 and 10) for a convenient visual presentation of expected engine order excitation of blade resonances at various rotational speeds. Note that the minor effect at these speeds of increasing flexural frequency with RPM is shown.

Interference (or Campbell) diagrams plot blade natural frequencies versus rotor speed. On log-log paper, lines of unity slope represent multiples of the rotor rotational speed (of frequency of rotation) known as engine orders. Integer values of the engine frequency commonly cause high vibratory excitation, especially from low orders (1 through 4) and any order from a regular flow variation around the blade/nozzle circumference (such as nozzle passing

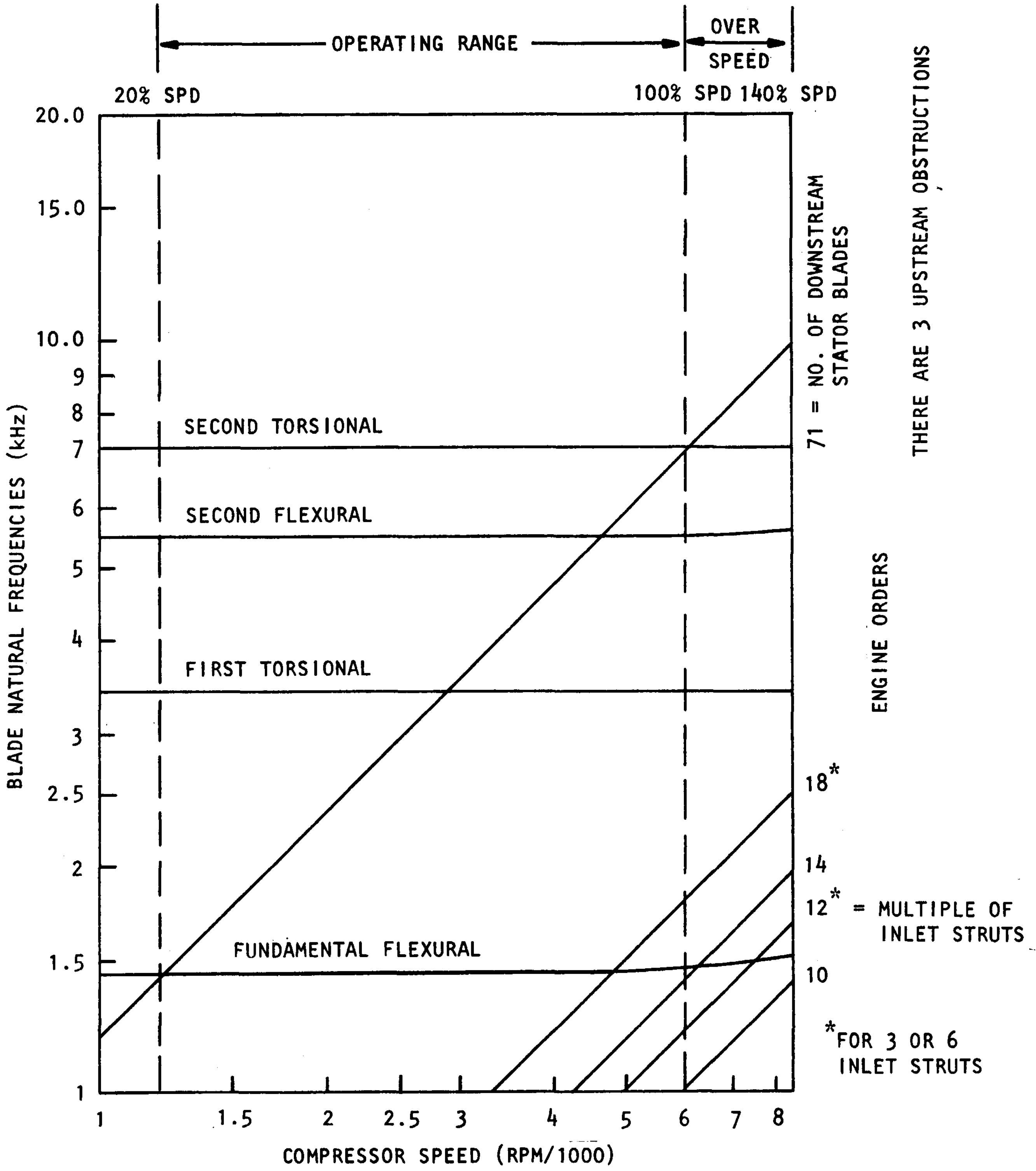


Fig. 9. GCFR circulator first stage axial compressor blade interference diagram

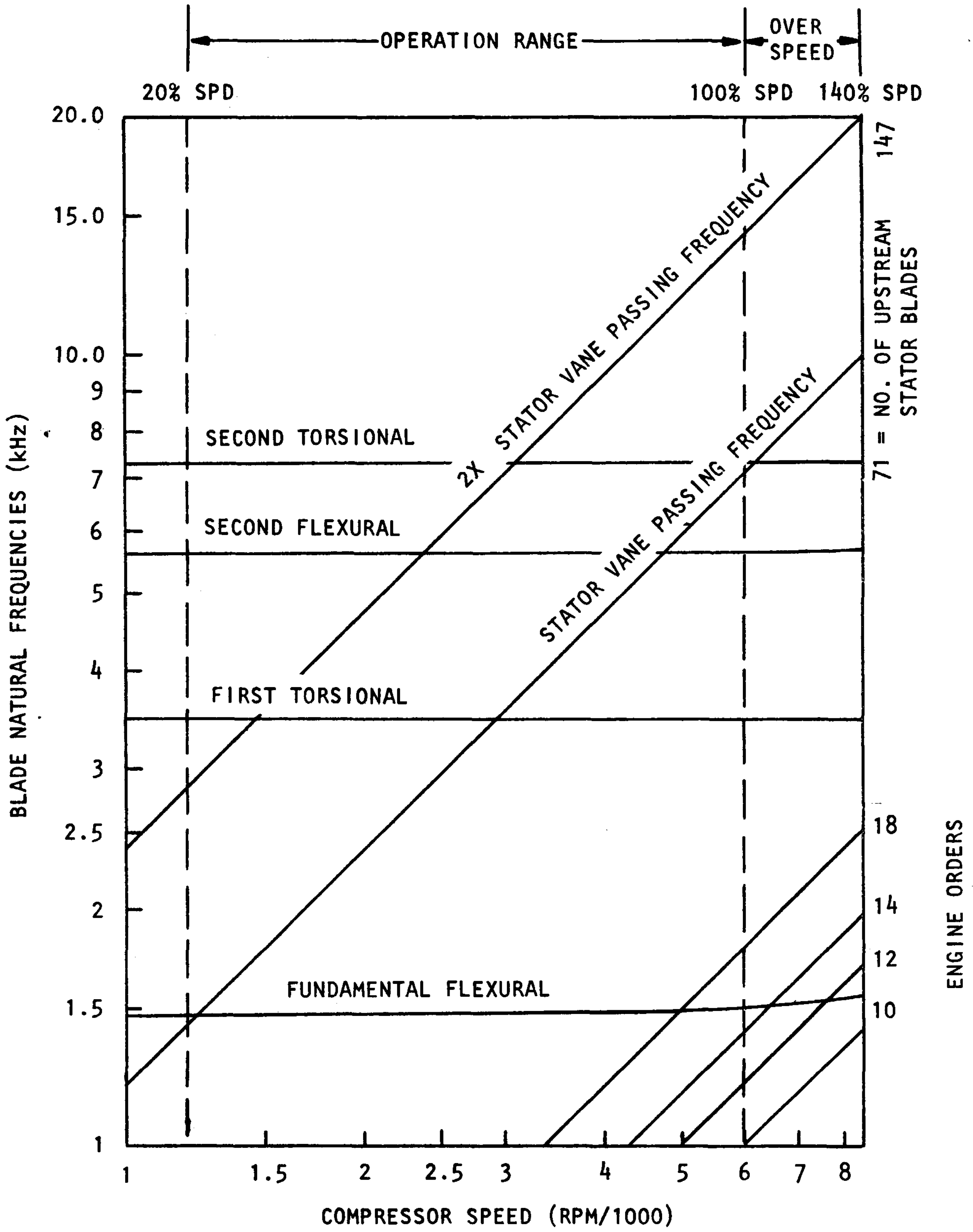


Fig. 10. GCFR circulator second stage axial compressor blade interference diagram

frequencies or their harmonics). Because destructive blade vibration often occurs at such integer engine excitation, it is necessary to examine for potential troubles early in the design phase and to make appropriate remedies before approving a prototype. Such an examination follows for each row of the circulator rotor.

Compressor stage 1 blades (Figure 9) will experience engine order excitation mainly from inflow non-uniformity (not determined, but most probably of order 4 or less) and cross duct inlet strut wakes (either 3 or 6 in number). From the diagram it can be seen that the only mode crossing harmonics of those strut disturbances is the fundamental flexural (or first flap) mode. Because only harmonic orders (lower level components of the Fourier series describing strut wakes) cross the first blade mode these excitations should not cause blade failure from excessive alternating stresses at resonance.

71 downstream stator vanes will likewise cause small excitation upstream on these blades and the dashed line representing engine order (E.O.) #71 is shown for interest only.

Compressor stage 2 blades (Figure 10) are sufficiently similar to stage 1 blades so that their computed natural frequencies are close to one another. The inlet strut wakes will be experienced by this second stage as well but because of greater distance and flow through stage 1 these wakes will have a lesser effect on this stage.

Of greater concern on stage 2 is the line (E.O. #71) of stator vane passing frequency caused by the 71 stator vane flows impinging on the first torsional mode at about 3000 RPM (50% speed), on the second flexural mode at 4700 RPM (78% speed) and on the second torsional mode at about 6000 RPM (100% speed).

Because aerodynamic differential pressures across the blade airfoils vary directly with the square of the rotor speed the input (forcing function) at 3000 RPM to the first torsional vibration is about 1/4 of what it would be at 100% speed. Therefore, the input to the first torsional at 50% speed and the fundamental flap at the still lower 20% speed are at favorably reduced levels. Whether those speeds will prove detrimental to blade life seems doubtful, but only further analysis and actual rig testing can establish design adequacy.

Stator vane passing frequency crossing the second flexural at 78% speed and the second torsional at 100% speed will have higher excitation pressures. However, because of the counterflexure less effective blade surface area is available to drive the blade at these frequencies. That is, while a major portion of the surface is in phase with the forcing (pressure) fluctuation and therefore receiving vibration energy from it, other areas are in opposite phase and are giving up vibration energy (damping). Also material damping is improved over first modes since stress maxima occur at other than the blade root, causing higher hysteresis levels at more locations in the blade. The combined effects of higher excitation pressures with less effective area and more damping would indicate that these upper modes may have stress maximums no worse than the first torsional mode discussed previously. Comments about rig testing and/or further analysis are also applicable to these upper modes.

4.3 Steam Turbine Design

The circulator is driven by a two stage, axial flow steam turbine which is operated on superheated steam in series with the main power plant turbines. The speed of this superheated steam turbine is controlled by the regulation of steam flow. The cycle is inherently self-regulating because the cooling requirements of the reactor plant are directly related to the supply of steam produced. Therefore, under normal operation the steam energy available for driving the circulators is adequate since a 7% power margin is provided in the superheated steam supply.

4.3.1 Steam Turbine Calculation Method

The computer code DSTURB, which calculates a complete axial flow steam turbine blading design when given an initial set of conditions, was used to establish the axial flow steam turbine configuration. The compressor power requirements determined the basis for the computer study.

The DSTURB code is similar to the AXCOMP code in that:

1. Every segment along the radius of the blade must be capable of extracting equal work from the steam and maintaining a constant axial through-flow velocity for every segment along its radial length.

2. The steam has no radial velocity during its traversal of the blade.

The first step in the design process is to determine the enthalpy drop per stage from the input values of specific work and the number of stages. With this information, the cross sectional annulus area for each stage of the turbine is calculated. This is done with four options for the change of diameter: (1) constant outer diameter, (2) constant mean diameter, (3) constant inner diameter, and (4) constant blade height.

The blade geometry, including camber, stagger angle, solidity, and blade profile, can then be determined. Various loss calculations are built into the program.

4.3.2 Turbine Aerodynamic Design

In order to obtain high turbine efficiency, a vortex free blade design was chosen for the rotor. This design has zero, or very low, exit whirl losses. This is accomplished by designing blades which have axial flow velocity out of any blade cross section from the hub to the tip. As a consequence of free vortex, equal work should be extracted at each radial section. The velocity triangles should be determined accurately owing to changes in axial velocity from inlet to outlet caused by density changes. The blade sections at the hub are nearly perfect impulse blades, whereas the blades near the tip result in about 12% reaction. In contrast, a pure impulse turbine would have a slightly smaller tip diameter, a longer blade, and lower efficiency with resulting low turbine exit pressure. The relationship between the amount of reaction in the turbine, the efficiency, and the blade height is clearly in favor of the partial reaction turbine, as demonstrated by the normalized chart shown in Figure 11. For a fixed speed, a large diameter, high reaction, and short blade are necessary in order to obtain high efficiency.

The design parameters for the circulator drive turbine were selected from an optimization study of the overall turbine efficiency, hub diameter blade height, blade camber, axial flow velocity, blade tip speed, flow angles, etc. (Figures 11 through 16). The conditions shown in the figures are different from the reference design but are included to show the trend in parameters.

The chosen parameters were:

Turbine speed	6000 rpm
Hub diameter	609 mm (24.0 in.)
Axial flow velocity	92 mm/sec (300 ft/sec)
Turbine overall efficiency	82.0%
Mean reaction	6%
Blade height	14 mm (.53 in.)
Blade camber	5.2/3.6

To obtain higher efficiency would require a larger hub diameter and consequently a shorter blade height. A lower axial velocity would increase the camber above 6 which makes it difficult to design the blade installation on the rim. 6000 rpm was found to be the highest speed which would yield good flow angles. High rotating speeds are desirable in order to obtain minimum overall circulator dimensions.

The final steam turbine design parameters are listed in Table 1. The turbine performance characteristics are presented in Figure 6, which includes normalized turbine enthalpy drop versus steam flow rate for various aerodynamic speeds.

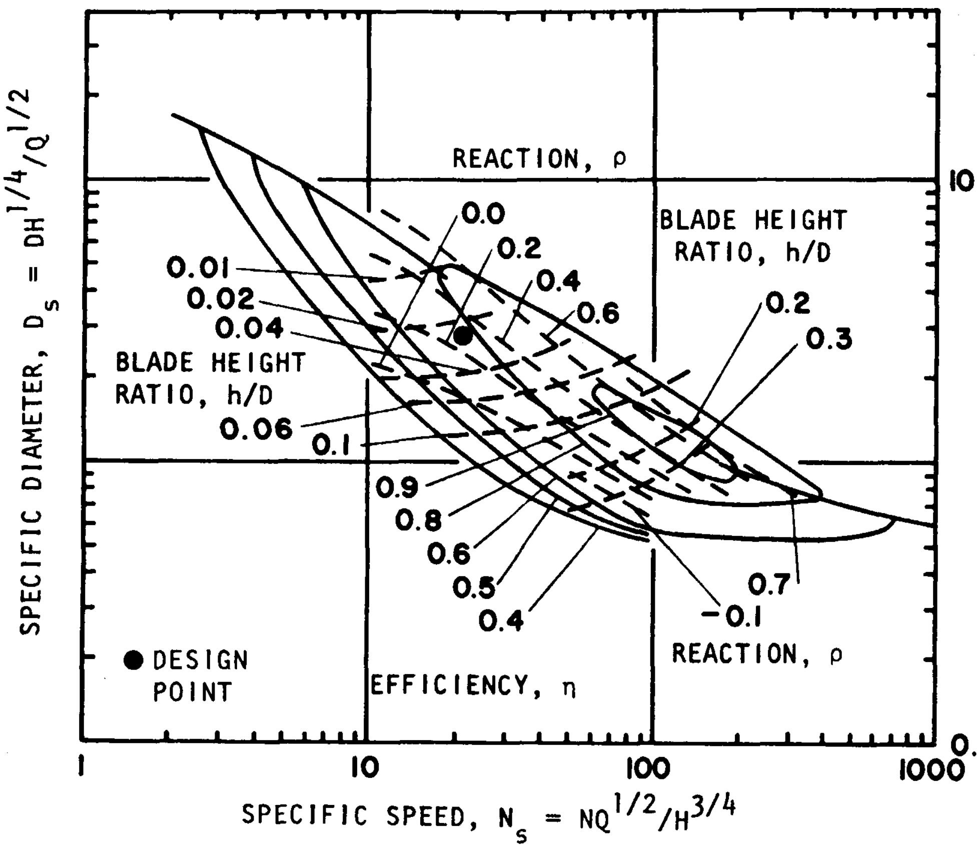


Fig. 11. Basic full admission diagrams showing performance and basic geometry

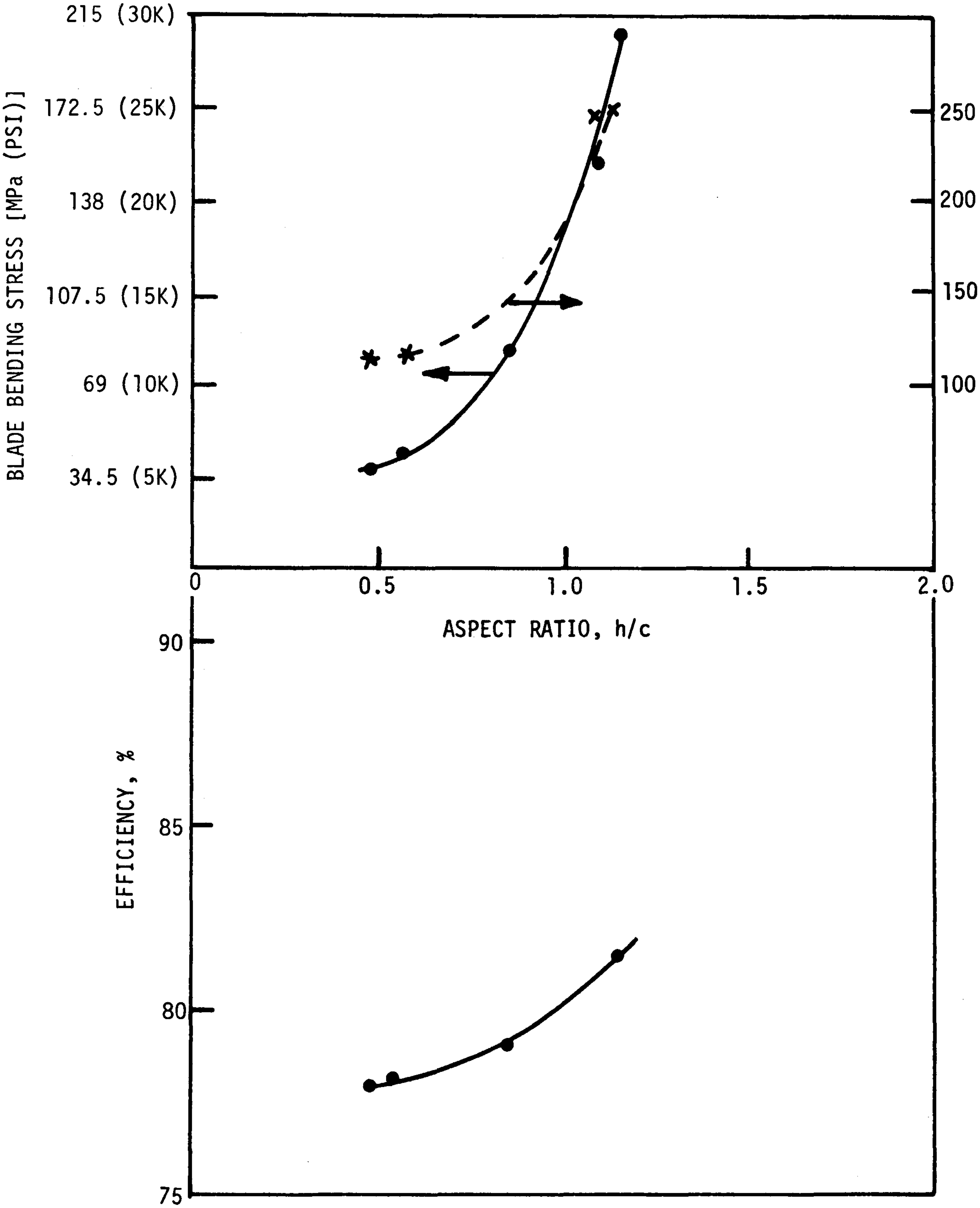


Fig. 12. Blade bending stresses and turbine efficiency as functions of aspect ratio

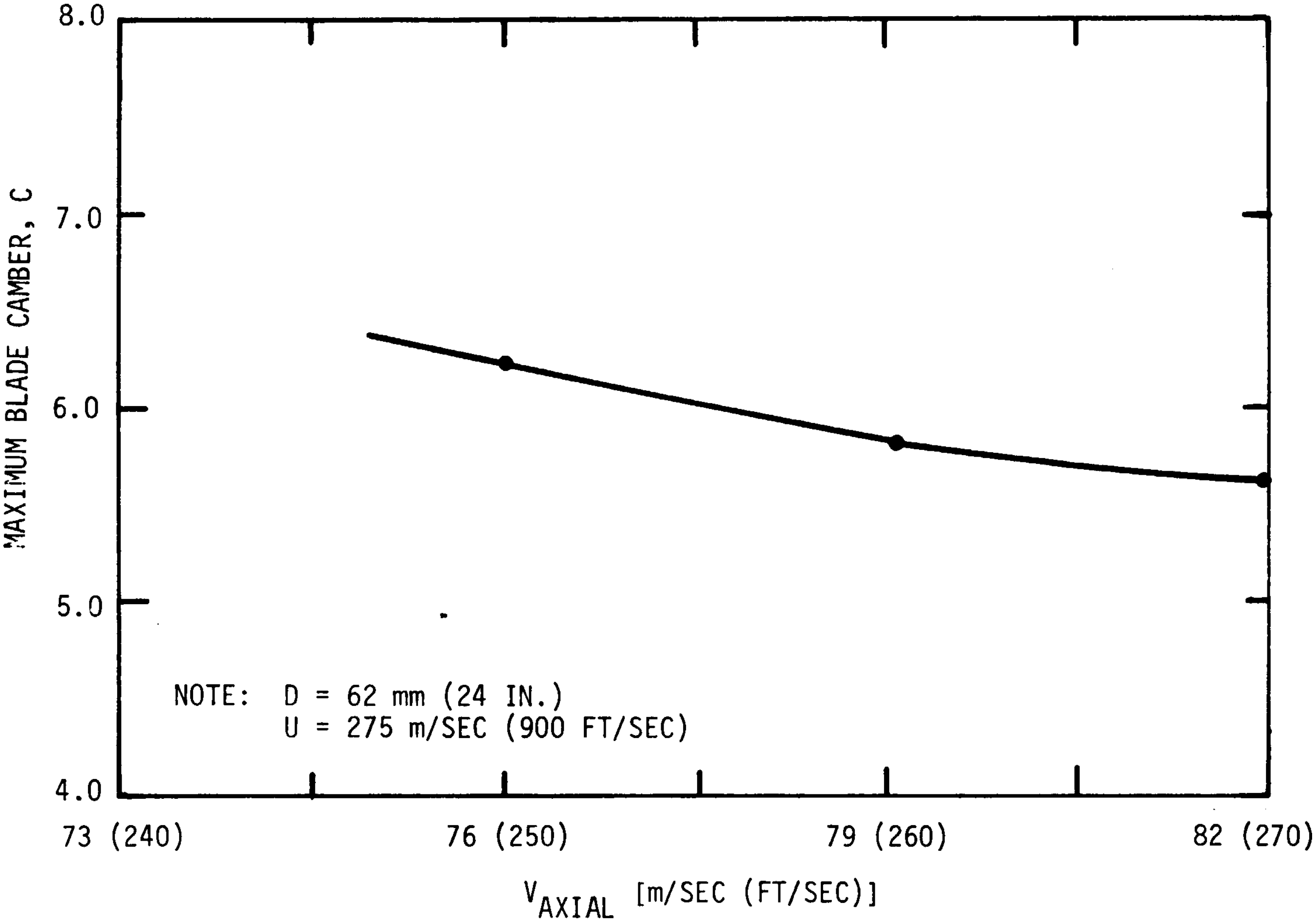


Fig. 13. Influence of axial velocity component on maximum rotor blade camber

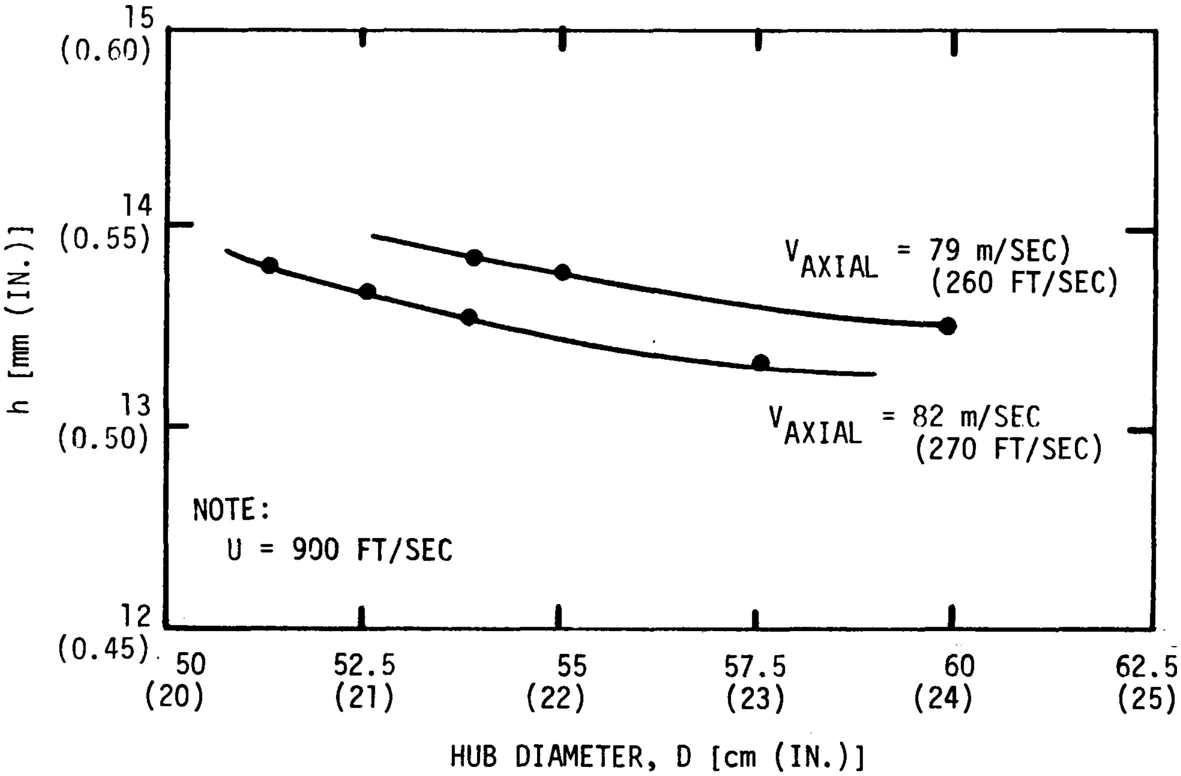


Fig. 14. Influence of hub diameter and axial velocity on blade height

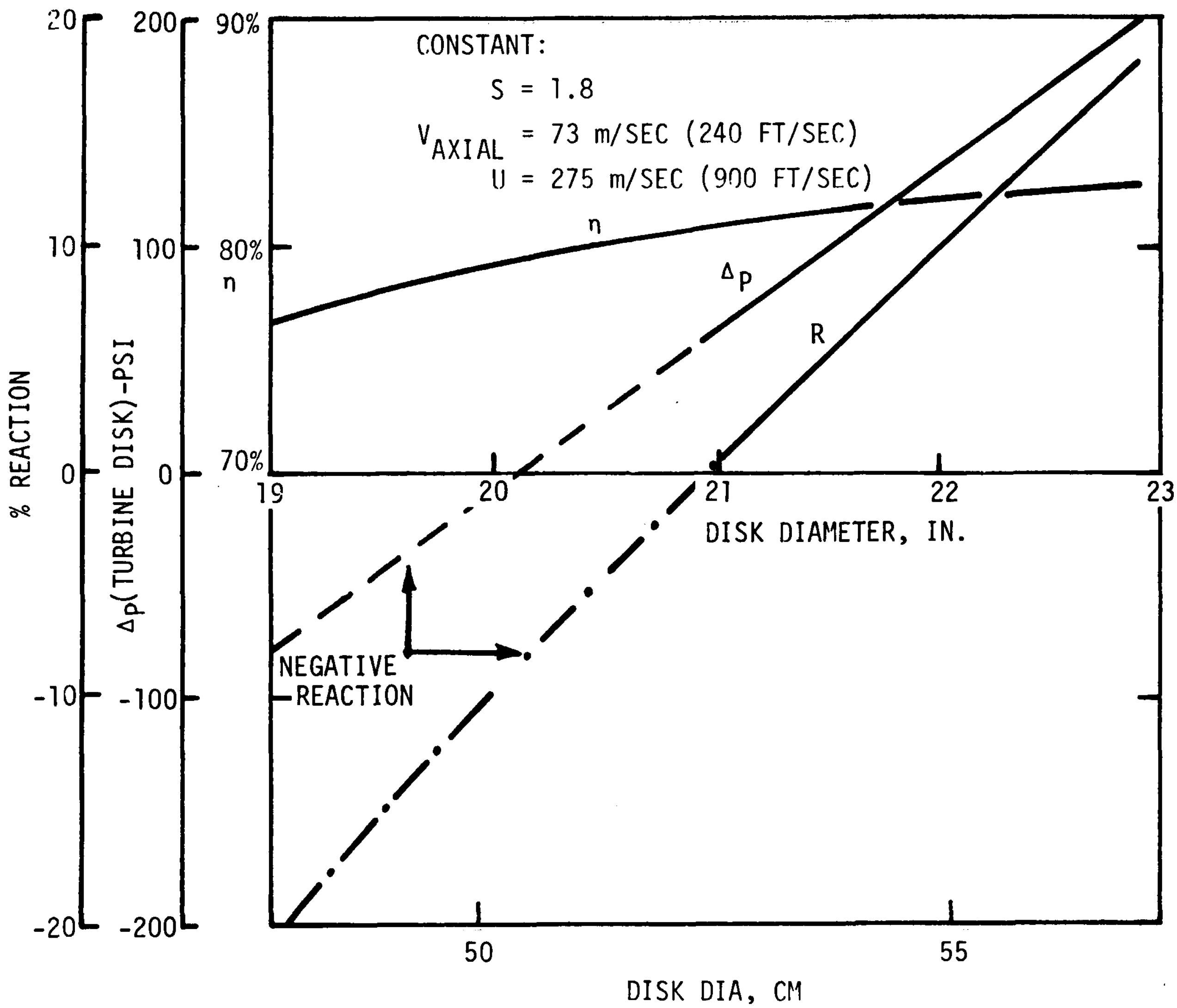


Fig. 15. Efficiency, reaction and ΔP across turbine disk as function of disk diameter

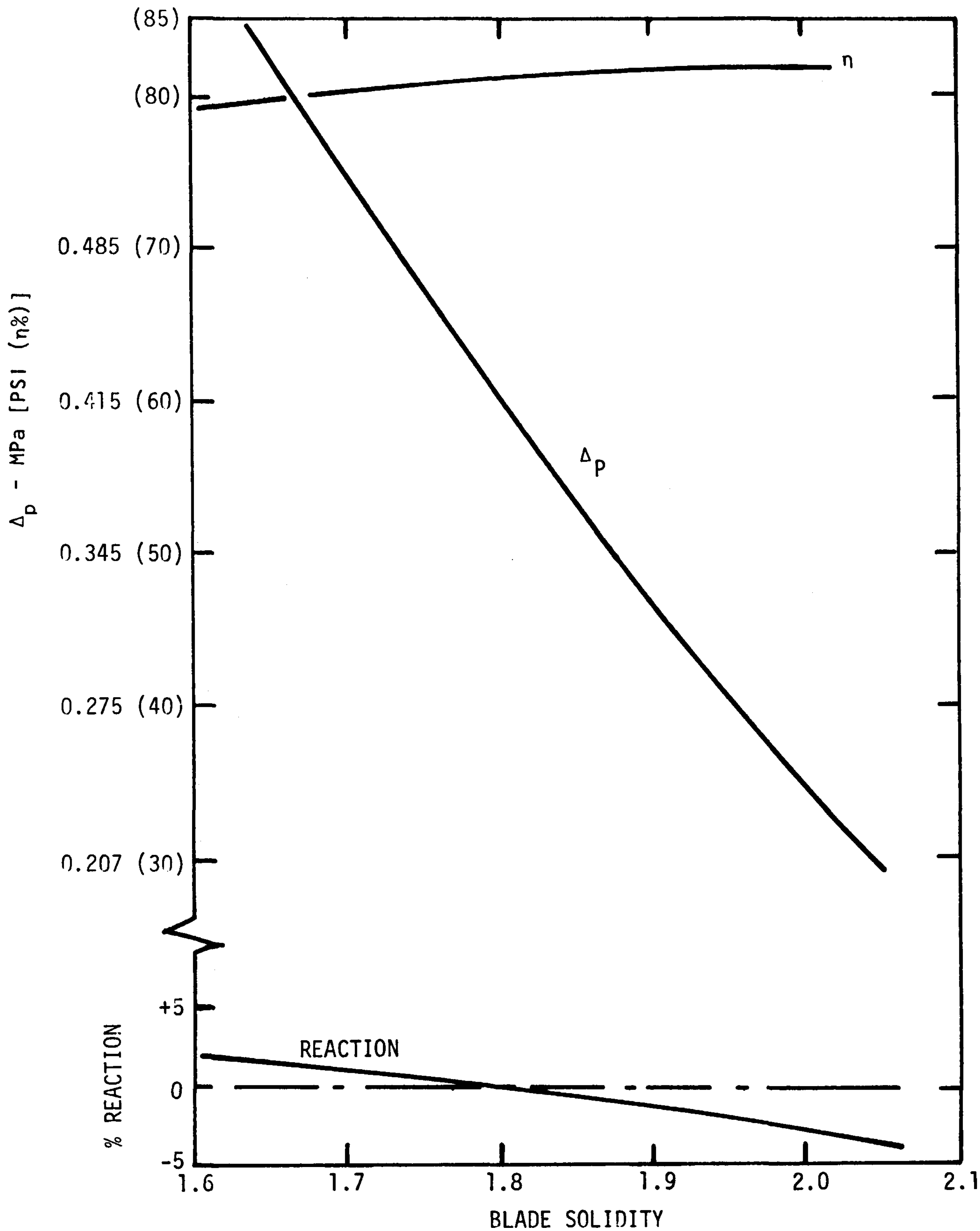


Fig. 16. Efficiency, ΔP across disk and percent reaction as function of blade solidity

4.3.3 Restrictions on Circulator Turbine Design Parameters

The GCFR cycle requires large helium volume flows with high compression (> 1.04) through the circulator on one side of a common shaft and with low steam volume flows and high ΔH (or power extraction close to the limit for a single stage) on the other end of the shaft. These conditions impose the following limitations on the circulator design.

Compressor

- High tip speed $\sqrt{\Delta P}$
- Large compressor diameter $\sim V_{tip}/N$
- High disk rim stresses $\sim V^2_{tip}$
- Diameter is also limited by d, a, σ_{freq}.

Turbine

- Low RPM fixed by compressor
- Large diameter fixed by compressor
- Short blades fixed by D, N and W_{steam}
- Low aspect ratio fixed by ΔH and σ_{freq}.
- High nozzle Mach number, $\Delta H = f\ (\Delta P,\ W_{He},\ 1/W_{steam})$, $M = C\sqrt{\Delta H}$
- High steam exit pressure = f (inlet pressure, ΔH)

Percent Reaction at Hub - 0%

It is necessary to use nearly pure impulse or 0% reaction at the hub in order to obtain minimum P across the disk and in turn obtain reasonable axial thrust loads on the circulator shaft. This is a very critical point because of the high steam inlet pressure of 2400 psi and the high H required.

In order to match the circulator and the drive turbine operating parameters it is necessary to design the turbine to operate at less than optimum design conditions. Also, the high steam pressure, low volume flow and high enthalpy drop severly restrict the turbine design parameters for a single stage turbine design. Therefore, a two stage turbine design was chosen. The turbine design is within the state-or-the art of steam turbine design, but for some parameters minimum acceptable values have been chosen to obtain a turbine which will satisfy all the required operating conditions. Some of the critical parameters and the reasons for chosing them are listed in Table 6.

4.3.4 Turbine Blade Vibrations

The turbine impulse blades are rather short (14 mm, 0.531" long) with low aspect ratio (chord $\sim$ 25mm, 1.0" wide). Consequently, the fundamental frequencies are high (Figures 17 and 18). The only interference with engine order lines of possible significance are the nozzle passing frequency and its first harmonic on the first torsional and first flexural modes. This significant E.O. approaches or crosses the natural frequency lines at 140% engine speed. Significant resonance may then occur at such overspeed conditions. Twice the nozzle passing frequency does cross the first torsional mode at 68% speed. Whether the nozzle passing frequency harmonic is strong enough to cause blade troubles at these speeds or not would involve further study and/or testing. The overspeed crossings, if the turbine is to function for extended periods of time at overspeed, could be avoided only by lowering the number of nozzles to less than 100 or possibly raise blade frequencies by increased taper ratio (thicker root and thinner tips) or decreased aspect ratio (causing undesirable aerodynamic effects).

4.4 Circulator and Turbine Flow Paths

The flow paths for both the circulator and drive turbine consist of an inlet to the blading and a diffuser following the blading. The main design effort has centered on the compressor helium flow paths since, as stated previously, pressure gradients and boundary layer characteristics are more critical during the compression process.

TABLE 6

RESTRICTIONS FOR CHOSEN DESIGN PARAMETERS FOR GCFR CIRCULATOR TURBINE

1. Blade Height - 14 mm (0.53")
   Restricted by small steam flow, tip speed required to obtain Δh and matching circulator speed and ΔP, large diameter required for outlet duct to clear bearing housing, obtain low centrifugal blade stresses. Approximately 12 mm (0.50") is the minimum recommended in the literature for large turbines.

2. Number of Blades - 113
   Result from requirements for low bending stresses of 6,000 psi (41.5 MPa) with a minimum aspect ratio of 0.50. More blades results in higher aspect ratio but also higher bending stresses.

3. Aspect Ratio - 0.53
   This is the minimum aspect ratio recommended in the literature, but is also the maximum allowable to obtain low bending stresses of 6,000 psi. Aspect ratio above 1 is desirable but not obtainable.

4. Axial Velocity - 300 ft/sec (92 m/sec)
   Lower velocities result in blade cambers higher than 6.0, which is undesirable for blade aerodynamics and installation problems.

5. Blade Bending Stresses - 5400 psi (37.2 MPa)
   This is the minimum stress level obtainable in order to satisfy the limitations on aspect ratio and blade height. It is also the maximum allowable in order to control blade excitation frequencies.

6. Blade Chord - 25 mm (1.00")
   Determined by blade height and aspect ratio limitations.

7. Efficiency - 82%
   Turbine efficiency is limited to 82% in order to satisfy other requirements above.

8. Percent Reaction at Hub - 0%

It is necessary to use nearly pure impulse of 0% reaction at the hub in order to obtain minimum ΔP across the disk and in turn obtian reasonalbe axial thrust loads on the circulator shaft. This is a very critical point because of the high steam inlet pressure of 2400 psi and the high ΔH required.

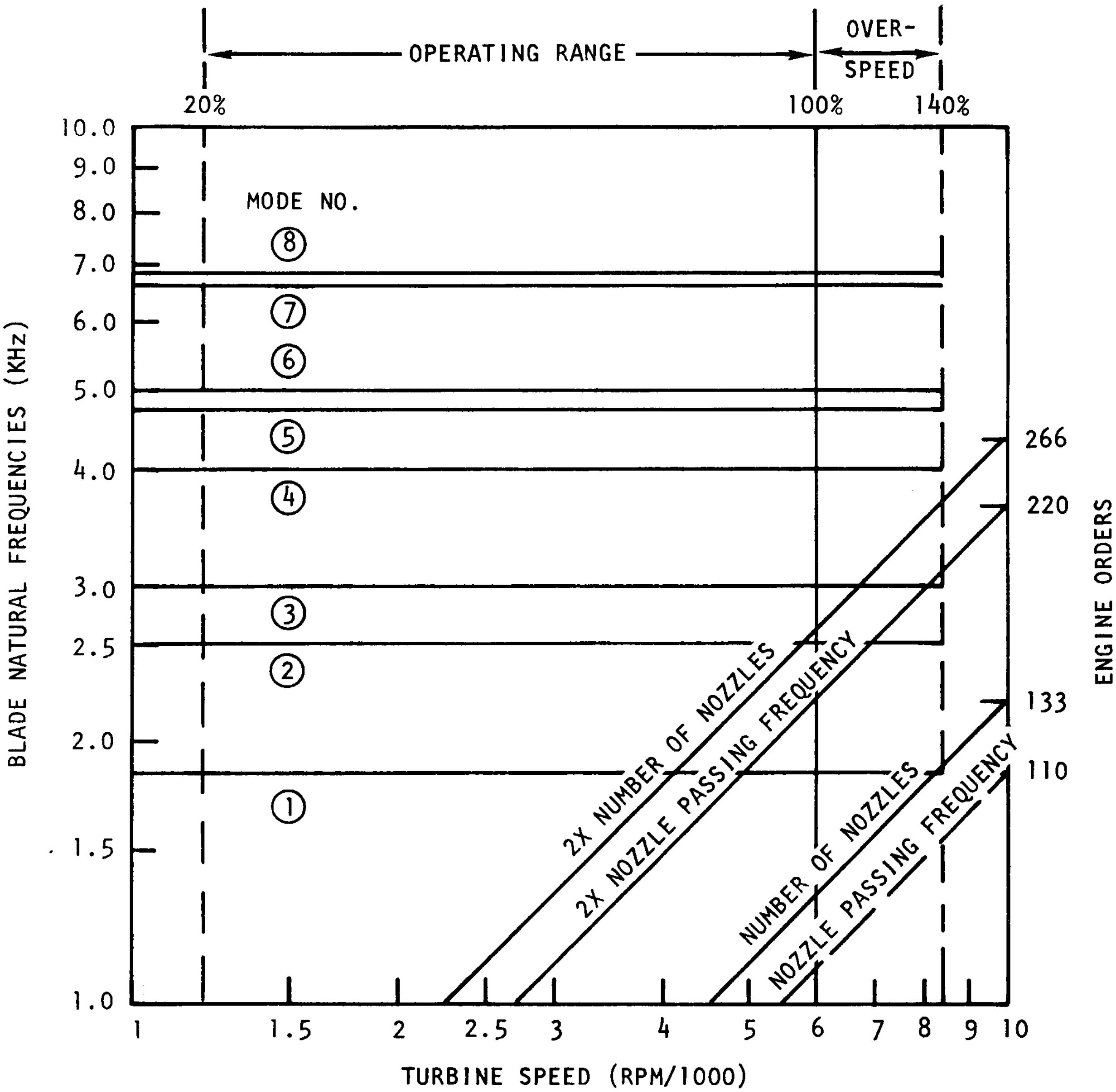


Fig. 17. GCFR circulator first stage turbine blade interference diagram

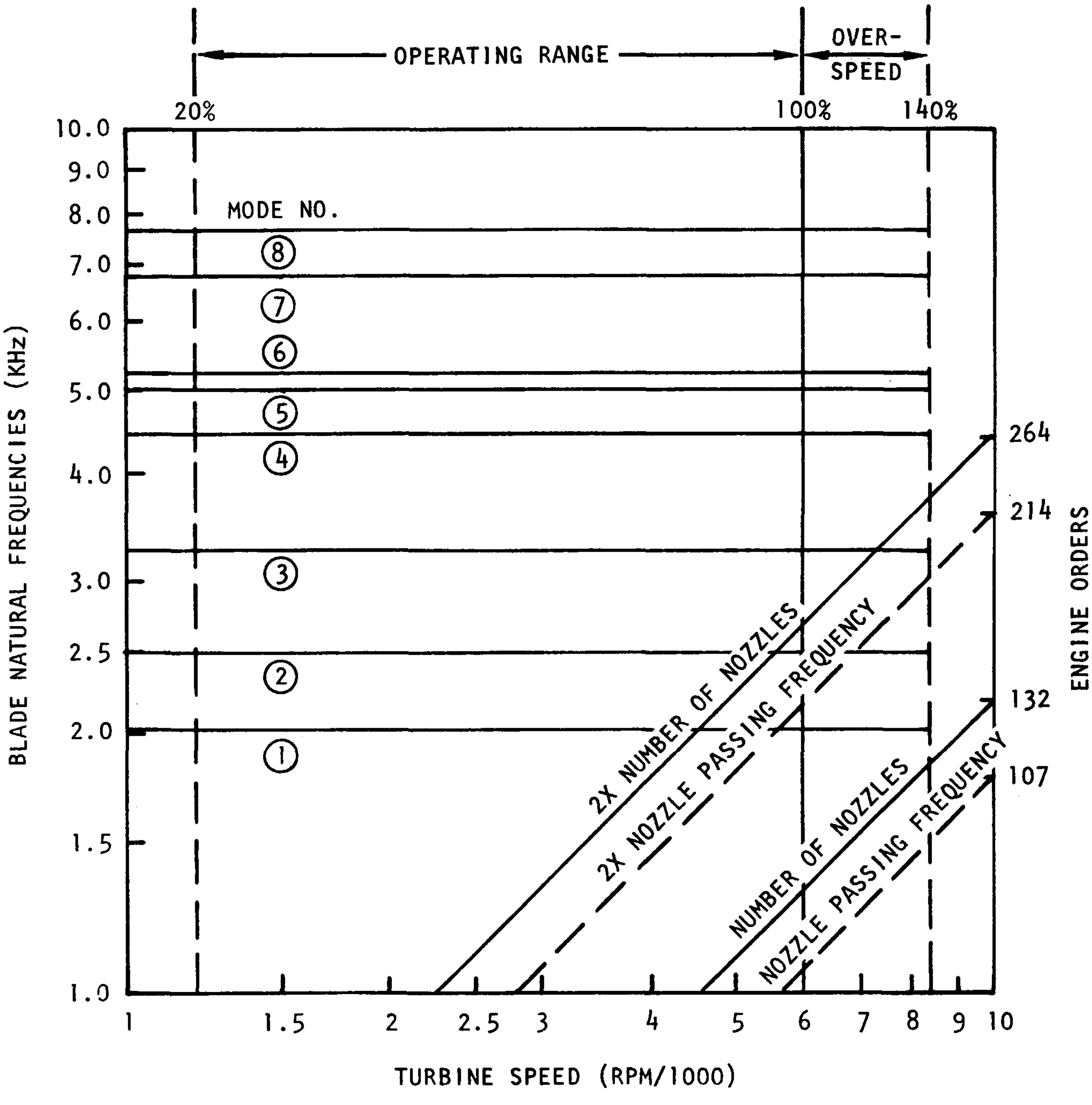


Fig. 18. GCFR circulator second stage turbine blade interference diagram

4.4.1 Inlets

The primary function of the circulator inlet is to deliver the helium flow axially into the rotor blades with minimum momentum and frictional losses and also with a uniform velocity profile. To accomplish this, it is desirable to accelerate the helium at a reasonable rate within the turn as the rotor blades are approached. An ideal process would result in a monotonic static pressure decrease as the helium velocity increases. Since the fluid at the blade tip has to be turned more quickly, the flow in this region has a tendency to separate from the inlet contour. For this reason the design effort has been concentrated on the blade tip flow contour.

A secondary function of the circulator inlet is to serve as a flow-monitoring device within the reactor helium circuit. For this application the inlet can be compared to a nozzle for which, when properly calibrated, a flow coefficient can be determined. Then the differential pressure loss measured across the inlet can be used as an indication of the flow passing through the compressor. This flow indication is an important parameter for determining satisfactory compressor performance during operation of the reactor.

The inlet design of the helium circulator is subject to the following restrictions:

1. The helium flow must be turned from the steam generator discharge plenum into the compressor rotor blades.
2. Dimensional limitations are imposed by the compressor bearing configuration and the reactor arrangement.
3. Support struts must pass through the inlet.
4. The innermost segment of the inlet, along with the compressor blading, must be removable from the reactor through the circulator access penetration.

The blade tip flow contour is considered to be the most important region of the inlet owing to the large directional change of the flow. Flow separation may be encouraged by this directional change, but the fact that the flow is being rapidly accelerated will tend to inhibit separation.

By definition, the flow separation on any of the inlet contours would be initiated by the detachment of the boundary layer as the flow progressively loses energy in overcoming the surface drag. Accelerating flow is less likely to separate than decelerating flow because the drag forces decrease in proportion to the lower density. Separation can also be prevented or delayed by continually keeping a high energy level in the boundary layer. However, for the GCFR case the use of auxiliary control methods is not considered necessary.

The turbine inlet flow is less critical and the geometry is also simpler. Boundary layer separation is not an important consideration because of the positive surface pressure gradient associated with flow through the turbine. Therefore, a gradual reduction in flow area, providing a smoothly accelerated flow, for the transition from the inlet pipe to the full admission annular nozzle blading is sufficient for the final turbine inlet design.

4.4.2 Diffusers

The function of the compressor diffuser is to attain maximum static pressure recovery as the helium flow is decelerated from the compressor stator blade discharge into the reactor plenum. The diffusion process must be rapid enough to limit friction, momentum, and drag losses, but at the same time must be smooth and uniform to inhibit flow separation from the wall surfaces.

The circulator diffuser has gradually diverging walls formed by the tapered center body which equals a diffusion ratio of 3.5:1. The effective area expansion angle is a constant 16^{o} from the circulator exit to the center duct leading to the core (see Figure 19).

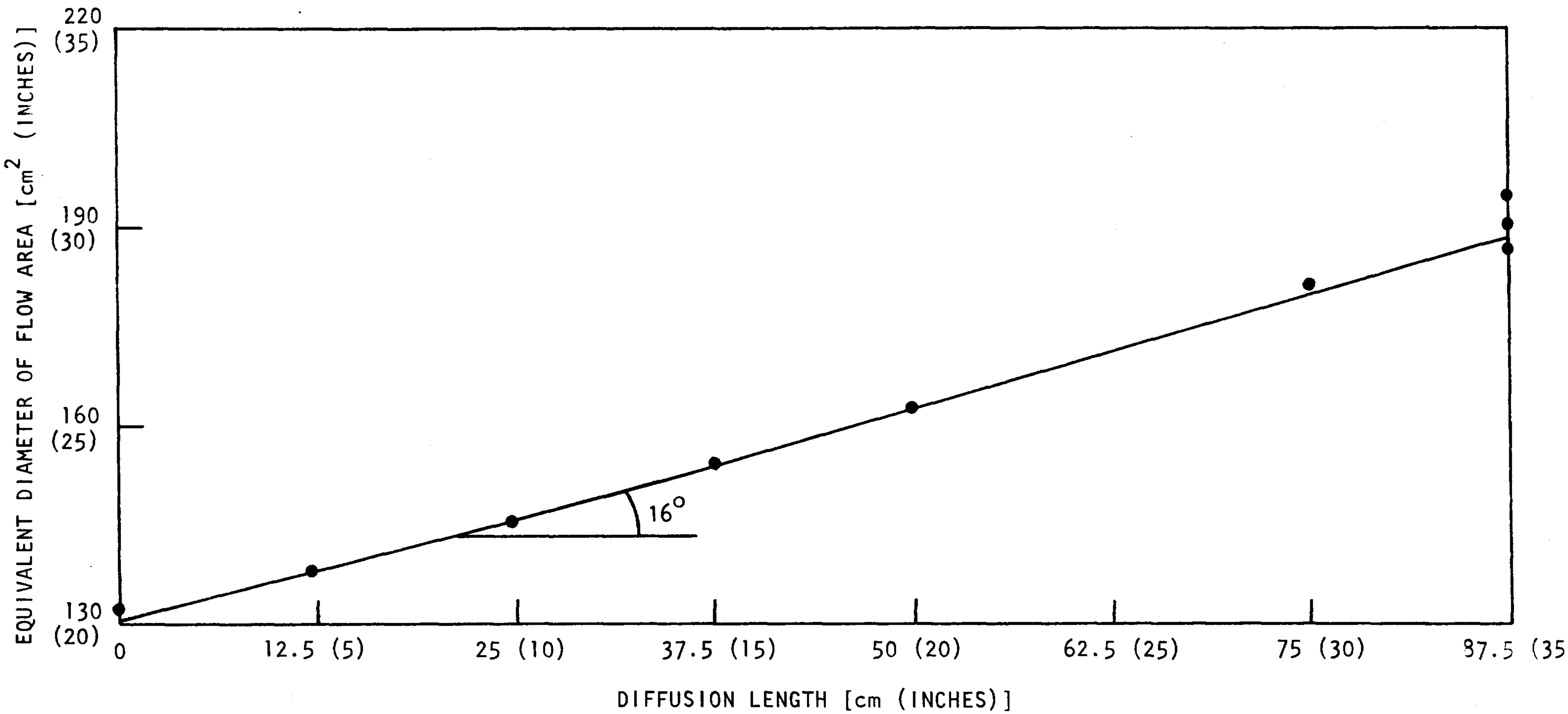


Fig. 19. Effective diameter vs circulator diffusor length

Expansion to a larger discharge area over a longer distance could be accomplished, but any additional pressure recovery would essentially be offset by the increased wall drag losses. However, the diffuser exit is extended on one side to minimize turbulence losses at the exit and to guide the flow toward the valve entrance, thus reducing the velocity head losses. The diffusion ratio at duct exit is 7:1.

The steam turbine diffuser has gradually diverging walls which provides the most efficient diffusion and is also a very compact design which minimizes losses due to wall drag. Also, the diffusion flow has a minimum tendency to separate at the boundaries in this type of diffusion.

5. MECHANICAL DESIGN

5.1 Rotor Disks

5.1.1 Computation Methods

The circulator and turbine rotor disks were designed with the aid of the computer code DISK (sample output is given in Ref. 1).

From the input variables (i.e., thickness, temperature, density, modulus of elasticity, Poisson's ratio, and expansion coefficient) and from the given boundary conditions, the code computes the stresses and displacements at a number of prefixed points along the radius as well as the disk weight and moments of inertia.

5.1.2 Disk Design

The profile selected for the circulator rotor disk is illustrated in Figure 20, which also contains a plot of the stresses at design condition. A similar plot of the turbine rotor disk characteristics is shown in Figure 21. The maximum design stress level is 35,000 psi (242 MPa), for the compressor rotor disk and 25,000 psi (173 MPa) for the turbine rotor disk.

5.1.3 Blade Root

The blade root was optimized using the DOVTAL computer program which calculates the root stresses and the disk rim stresses. The shank width, the blade root angle, the width of the blade root and the disk thickness above the blade root are some of the more important input variables. A wide root results in low blade root stresses but high disk rim stresses. A compromise will yield approximately the same stress levels in the blade root and in the disk rim. A narrow shank and a small blade root angle both yield low disk rim stresses.

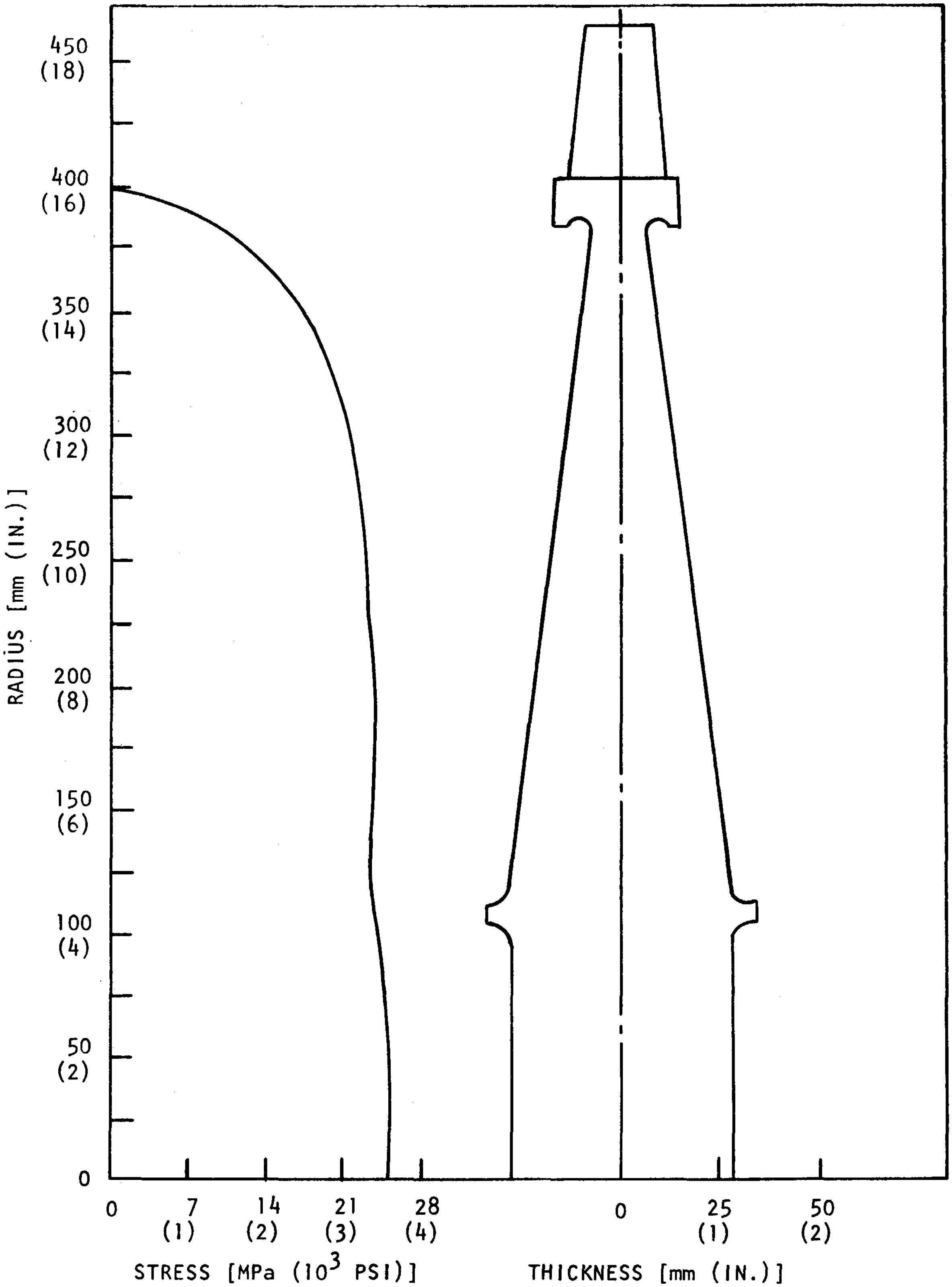


Fig. 20. Circulator disk thickness and stress

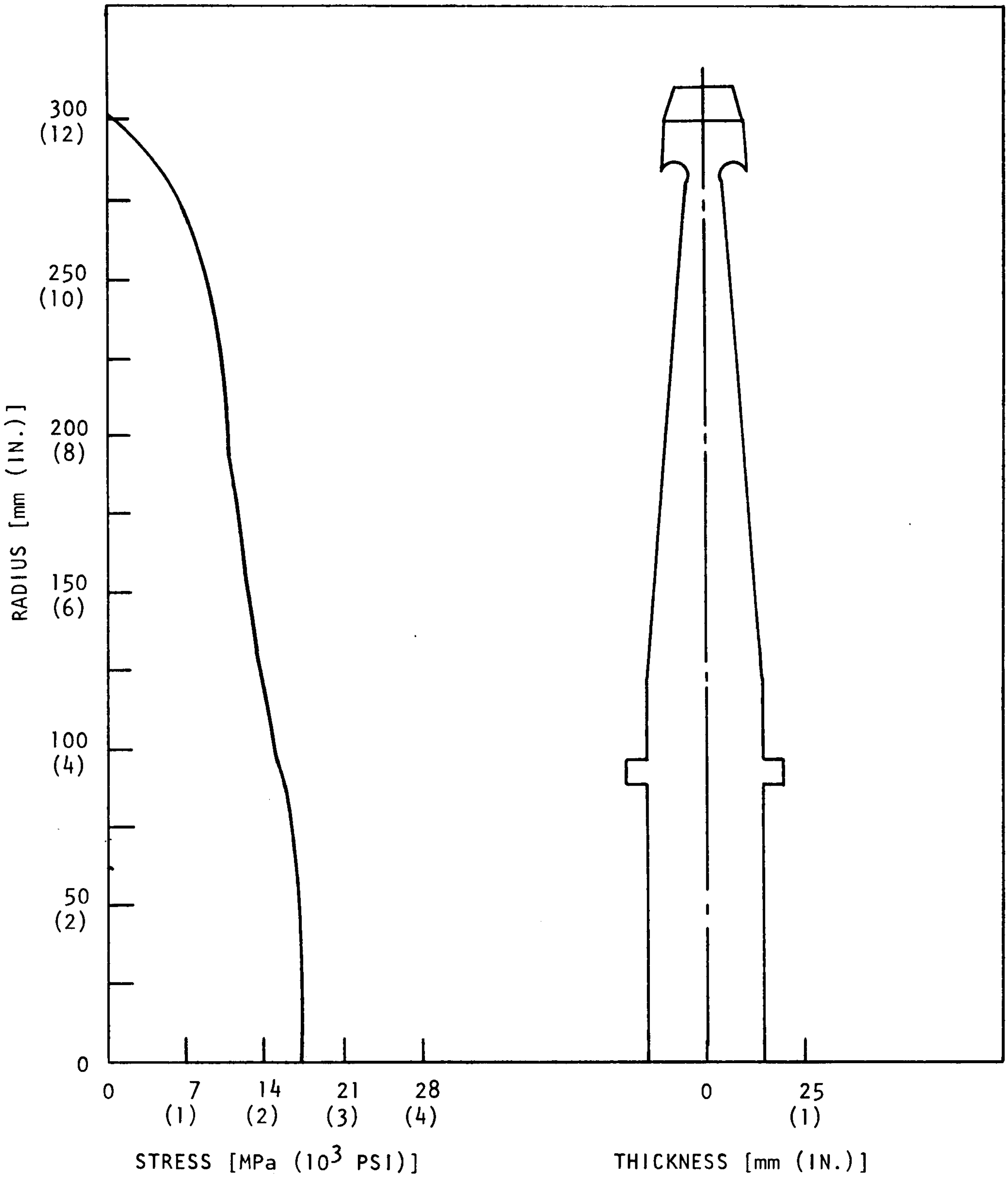


Fig. 21. Turbine disk thickness and stress

By varying all the input variables and plotting the resulting stress levels an optimized design was chosen to yield the best overall blade root design. Maximum stress levels at design operating speed will be less than 49,000 psi (276 MPa) which corresponds to a safety factor of 3.5 for 422 st.st. with a yield limit greater than 140,000 psi (965 MPa) at 650°F (344°C).

5.2 Circulator Bearings and Shaft

Water-lubricated journal and thrust bearings were chosen to:

1. Combine the lubrication and cooling functions.
2. Eliminate the need for oil-steam seals.
3. Eliminate the problem of oil compatibility with radioactivity.
4. Minimize the cost and complexity of the necessary auxiliary equipment.

Current experience at General Atomic with combination hydrodynamic and hydrostatic water-lubricated bearings that prevent contact between adjacent internal components has shown this to be a satisfactory concept well suited to the particular demands of this type of application.

5.2.1 Computation Methods

Computer programs have been used for the preliminary evaluation and sizing of the water-lubricated bearing configuration for the GCFR circulator. These programs can be used for pocketed multi-pad journal bearings and pocketed multi-pad thrust bearings.

Rotational speed, fluid viscosity, pocket geometry, desired clearances, and fluid pressure are among the input variables for the code. Bearing capacity, Reynolds number, and flow rates are calculated by the code for each operating point.

The computer codes have been used to analyze the critical speed of the circulator rotor assembly. Rotation at a critical speed occurs when a system of rotating masses experiences severe vibration caused by the frequency of the centrifugal forces acting on the masses coinciding with a natural frequency of flexural oscillation of the system.

By using the principle that at a critical speed the centrifugal forces generated by the shaft's deflections are in equilibrium with the elastic reactions corresponding to these same deflections, the RODYN code establishes an iterative process by which centrifugal forces are computed from previous deflections (arbitrary for the first step), and new deflections are in turn computed from these centrifugal forces. This process converges and leads to the determination of the first critical speed. It was found to occur at approximately 146% of design speed, well past the range of anticipated continuous operating speeds. To obtain critical speeds which are higher than normal operating speeds, it is necessary to use a 9-in. (230 mm) shaft diameter and large bearing pads with water pressures of 1000 psig (6.9 MPa) above containment pressure of 1250 psi in order to obtain high stiffness of the bearings.

5.2.2 Bearing Design

The basic circulator bearing requirements were established in conjunction with the thrust loads generated by the compressor and turbine.

A preliminary bearing design was based on a scaling of previous concepts. Computer codes were then utilized to refine and improve the geometry until a final design was selected. The codes were further used to determine the thrust bearing flow rates.

An integral part of the water-lubricated bearing system is its isolation from the main helium circuit by a flow of purified buffer helium through a labyrinth seal at the compressor end of the shaft.

This permits only purified helium to leak into the reactor. Some purified helium also leaks into the bearings and is continuously removed by an auxiliary system as the water is cooled and recycled. A small amount of bearing water leakage at the turbine end can be tolerated because of the compatibility between steam and water; therefore, no seal problems exist in this area.

Figure 22 shows the predicted operating characteristics of the main thrust bearing for a pocket inlet pressure of 1000 psig (6.9 MPa). The final thrust capability of the bearings was made compatible and coordinated with the predicted thrust loads anticipated from the final compressor and turbine designs. Also, the thrust bearings will be designed to counteract any severe accident load conditions without any direct metal contact between rotating and stationary components. The biggest loads on the main or reverse thrust bearings would develop in case of a sudden loss of steam pressure due to an upstream pipe rupture (between the valve and turbine) or a sudden loss of helium pressure (depressurization accident) because of the large shaft diameter and the large pressures on both the helium and steam sides.

5.2.3 Bearing Thrust Loads

An analysis was made of the net axial thrust loading on the circulator shaft for normal accident, start-up, and shutdown conditions. Four operating points were investigated corresponding to 100%, 75%, 50%, and 25% of plant operating power. From the circulator performance requirements established for each point by the AXCOMP computer code, the operating conditions in the compressor and turbine were determined with the help of the MATCH, AXCOMP, and DSTURB codes. Next, all the aerodynamic parameters that affect the thrust were evaluated and tabulated. The resulting net upward or downward axial thrust was then calculated for the four power settings investigated.

A large shaft diameter (approximately 230 mm or 9 in.) is required in order to obtain a high bearing stiffness and critical

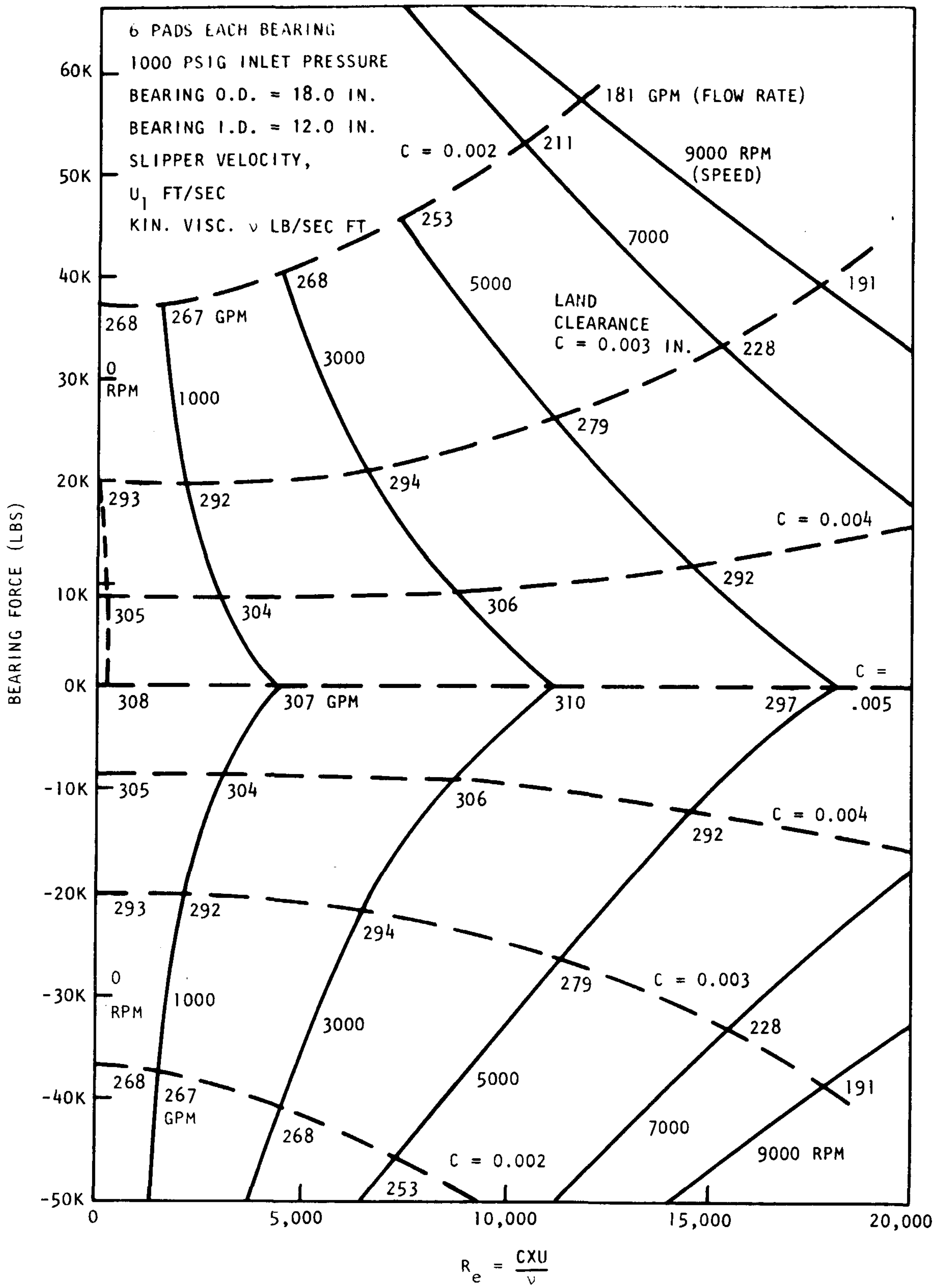


Fig. 22. Main and reverse thrust bearing characteristics

speeds above normal operating speeds. The resulting large cross-sectional area of the shaft, combined with a large helium pressure (approximately 1250 psi or 8.7 MPa) on the circulator end of the shaft and a large steam outlet pressure of 1800 psi (12.1 MPa) from the turbine nozzles will cause large net axial thrust loads during start-up, during shutdown, or in the event of a sudden loss of steam pressure due to an upstream pipe rupture (between the valve and turbine) or a sudden loss of helium pressure (depressurization accident). An analysis was made of practical means to reduce these large loads. To obtain reasonable net axial thrust loads during all normal and accident transient conditions, it was found necessary to introduce the high-pressure steam through the inside pipe of the two concentric pipes leading into and out of the turbine. It was also found that pressure equalization between the high- and low-pressure sides of both the circulator and the turbine disks would result in a higher net axial thrust for most normal and accident operating conditions than would a circulator without pressure equalization systems. This is partially due to the disk pressure loads and blade loads of the circulator being nearly balanced to the same loads on the turbine, whether or not a pressure equalization system is used on the circulator or the turbine. A pressure equalization system on only the circulator or the turbine would create worse thrust loads. Pressure equalization systems were thus proved to be undesirable for the 300 MW(e) circulators.

5.2.3.1 Load Analysis

The net axial thrust loading on the circulator shaft is the result of several combined loads on the circulator and turbine. Because of the vertical installation of the circulator in the reactor, there is a weight component on the shaft from the overall rotating assembly of the circulator and turbine. Then there is the thrust loadings on the circulator and turbine blades. There is also the pressure differential between the high- and low-pressure sides of the circulator and turbine disks acting on the entire disk surfaces except for the areas covered by the shaft. Finally, there is an unusually high axial thrust caused by high helium and steam pressures acting on a large cross-sectional area of the shaft. These complex loads were analyzed for various normal and accident operating conditions and to determine various practical design methods which could conceivably reduce the net axial loads on the shaft.

To calculate the net upward axial thrust, the static equilibrium of the shaft, as shown in Figure 23, requires that

$$F_T = -\tau_{1C} - \tau_{2C} + \tau_{1,T} + \tau_{2,T} - W_s + A_{1,T}\,\Delta p_{1,T} + A_{2,T}\,\Delta p_{2,T}$$

$$+ A_{1C}\,\Delta p_{1C} + A_{2C}\,\Delta p_2 + A_{shaft}\,p_{2C} - A_{shaft}\,p_{1,T}$$

where W_s = rotor weight,

$$A_{1,T} = \frac{\pi}{4}\,(D_{T2}^2 - d_2^2)\ ,$$

$$A_{s,T} = \frac{\pi}{4}\,(D_{T2}^2 - d_2^2)\ ,$$

$$A_{1,C} = \frac{\pi}{4}\,(D_{C1}^2 - d_2^2)\ ,$$

$$A_{2,C} = \frac{\pi}{4}\,(D_{C2}^2 - d_2^2)\ ,$$

$$A_{shaft} = \frac{\pi}{4}\,d_2^2\ ,$$

In the above formulae, subscripts 1 and 2 refer to the first and second stages and to inlet and outlet, respectively, C refers to the compressor, and T referes to the turbine. The symbol d_2 denotes the diameter of the water seal which separates the upper section of the bearing and seal system, balanced to the steam pressure $P_{1,T}$, from the lower section, balanced to the helium pressure $P_{2,C}$. The circulator is designed so that during normal operation, the pressure

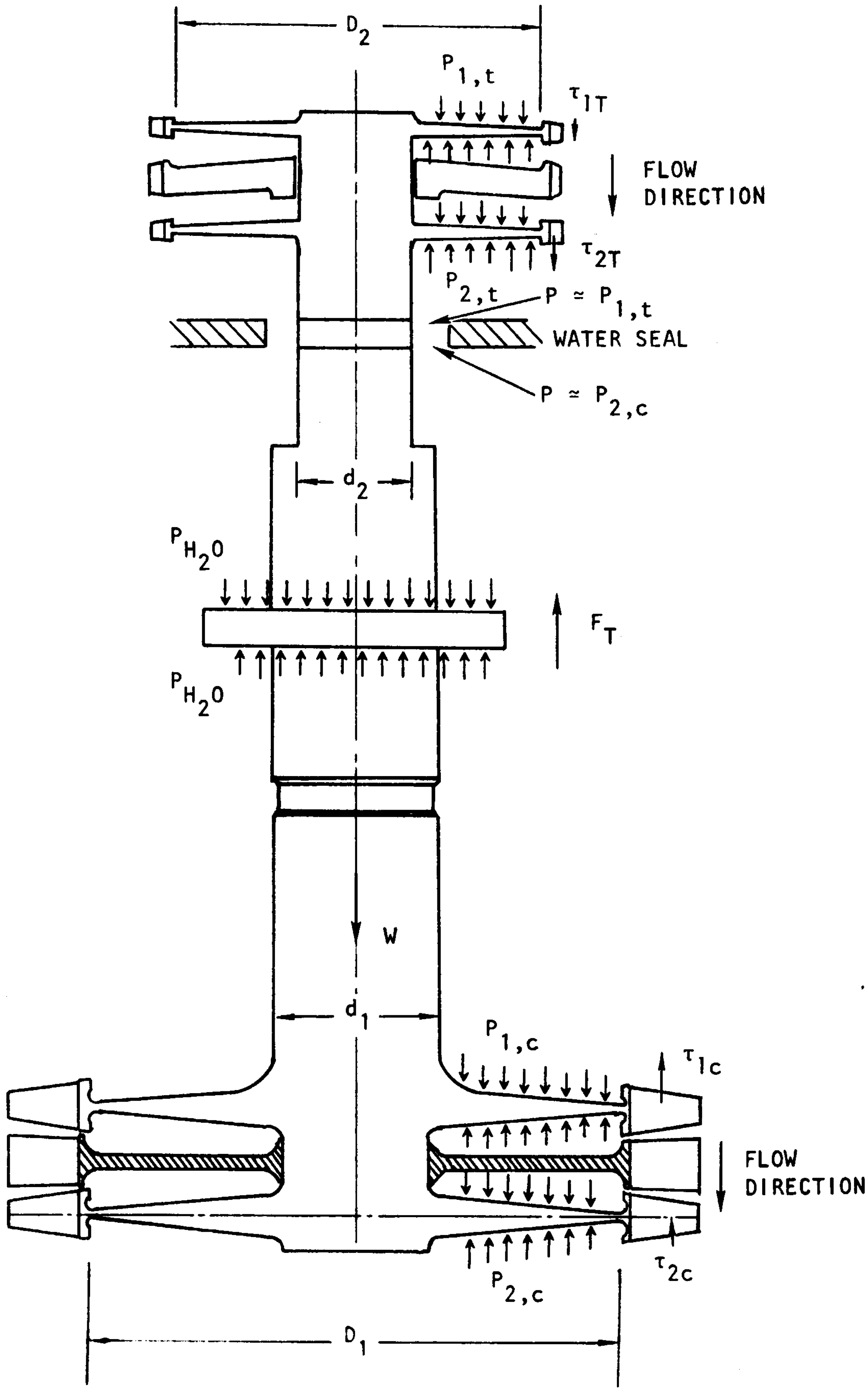


Fig. 23. Illustration of loads and dimensions of the rotating circulator components

loading on the steam side nearly balances the pressure loading on the helium side. The thrust loadings on the circulator blades (up) and turbine blades (down) as well as the total weight of the rotating components are included in the loadings described earlier.

Thrust During Start-up and Shutdown

The helium and steam pressures and the circulator speed (which is proportional to helium flow) during the start-up and shutdown transients can be obtained from the computer codes referred to earlier. At the low speeds typical of the start-up and shutdown transients, the blading thrusts and the pressure drops across the disks are very small. Therefore, during these transients the thrust practically reduces to the product of the shaft area times the helium-to-steam pressure differential. This differential is highest at full primary loop inventory and becomes small if the reactor is depressurized during the start-up or shutdown cycle. The largest thrusts thus occur at full helium inventory. These are as follows:

1. At the moment of start-up (zero speed), the thrust is about 33,000 lb (147,000 N) upward.

2. At the moment of shutdown (zero speed), the thrust is about 33,000 lb (147,000 N) upward.

3. During start-up and shutdown transients, the net axial thrust load will be less than the two previous cases because the steam pressure will counteract the helium pressure, thus lowering the axial thrust to between 33,000 lb (147,000 N) upward and 10,000 lb (44,500 N) downward.

Thrust During Upset and Accident Conditions

In addition to the normal operating conditions considered in previous sections, upset or accident conditions which could give rise to higher upward or downward thrust loads should be considered. The biggest loads on the main or reverse thrust bearings would develop in case of a sudden loss of steam pressure due to an upstream pipe rupture (between the valve and turbine) or a sudden loss of helium pressure (depressurization accident) because of the large shaft diameter and the large pressures on both the helium and steam sides.

Because of the magnitude of these loads, it is desirable to analyze practical means of reducing the loads as well as of determining the transient loads during normal operation for the methods investigated. The methods investigated were:

1. <u>High-Pressure Steam in Inside Pipe (τ positive)</u>

    a. Pressure equalization between the low- and high-pressure sides of the circulator disks ($\Delta p_C = 0$).

    b. Pressure equalization between the low- and high-pressure sides of the turbine disks ($\Delta p_T = 0$).

    c. A combination of items 1 and 2 ($\Delta p_C = 0$, $\Delta p_T = 0$).

    d. A system without pressure equalization as described in items 1 and 2.

2. <u>High-Pressure Steam in Outside Pipe (τ negative)</u>

    a. Pressure equalization between the low- and high-pressure sides of the circulator disks ($\Delta p_C = 0$).

    b. Pressure equalization between the low- and high-pressure sides of the turbine disks ($\Delta p_T = 0$).

    c. A combination of items 1 and 2 ($\Delta p_C = 0$, $\Delta p_T = 0$).

    d. A system without pressure equalization as described in items 1 and 2.

3. <u>Thrust Modulator using Steam Pressure Differentials</u>

    The accident conditions investigated for each of the preceding methods were:

    a. Depressurized DBA conditions.

    b. Upstream steam pipe rupture with total loss of steam inlet pressure.

TABLE 7

PRELIMINARY CIRCULATOR THRUST LOADS

FOR DESIGN POINT AND ACCIDENT CONDITIONS

Speed	Operating Conditions	Normal Net Thrust		Net Thrust With Thrust Modulator	
		(1000 lbs)	(1000N)	(1000 lbs)	(1000N)
100%	Normal	- 10	- 44	4	18
0%	Hot Start and Stop	33	147	33	147
100%	DBDA	- 27	- 120	- 27	-120
100%*	Upstream Pipe Rupture	83	370	83	370
100%	Downstream Pipe Rupture	-177	- 785	- 75	-333
133%	Downstream Pipe Rupture	-143	- 635	- 41	-182
- *	Casing Rupture	-265	-1180	10	44
133%	Bearing Cap. @ .0013 mm gap (.0005")	$\pm$300	$\pm$1330	$\pm$300	$\pm$1330
	@ .0025 mm gap (.0010")	$\pm$ 90	$\pm$ 390	$\pm$ 90	$\pm$390

*Affects only one circulator. Requires less than 1 mil bearing clearance.

c. Downstream steam pipe rupture.

The equations for the net axial thrust for each of the above methods can easily be derived from the thrust equation given earlier for the high-pressure steam in the inside pipe and for normal operation conditions. In all cases pressure equalization across the turbine and circulator disks were found to be of no significant advantage. A thrust modulator, however, was found to be necessary for accident conditions as described later. The thrust loads for design point and accident conditions are shown in Table 7. The same loads for variable speed conditions are shown graphically in Figure 24. Because of the large loads and narrow gap required of the thrust bearing during downstream pipe rupture accidents a thrust balancing system can be designed to give the loads shown in Figure 25.

In the depressurized DBA condition, the helium pressure in the reactor vessel is gradually reduced over several minutes. This "maximum downward thrust" case is considered at the end of this section.

An upstream steam pipe rupture with total loss of steam inlet pressure would result in loss of all steam pressure loads on the turbine blades, disk and shaft.

For a downstream steam pipe rupture, the upstream pressures and flow rates are controlled by the steam inlet valve and the turbine stator nozzles. Also, the turbine is nearly a pure impulse turbine, preventing any significant reaction and pressure drop across the rotor. A detailed analysis of this accident follows:

Down Stream Steam Pipe Rupture

A preliminary investigation of the effect of a down stream steam pipe rupture was undertaken to determine the effect on the two stage circulator series steam turbine drive. The results are shown in Figure 26. The steam flow rate increases until sonic flow occurs at the discharge from the second stage turbine nozzles. This condition limits the maximum flow to 314 lb/sec. (143 kg/sec). The turbine speed will increase to about 132% speed and the pressure drop across the

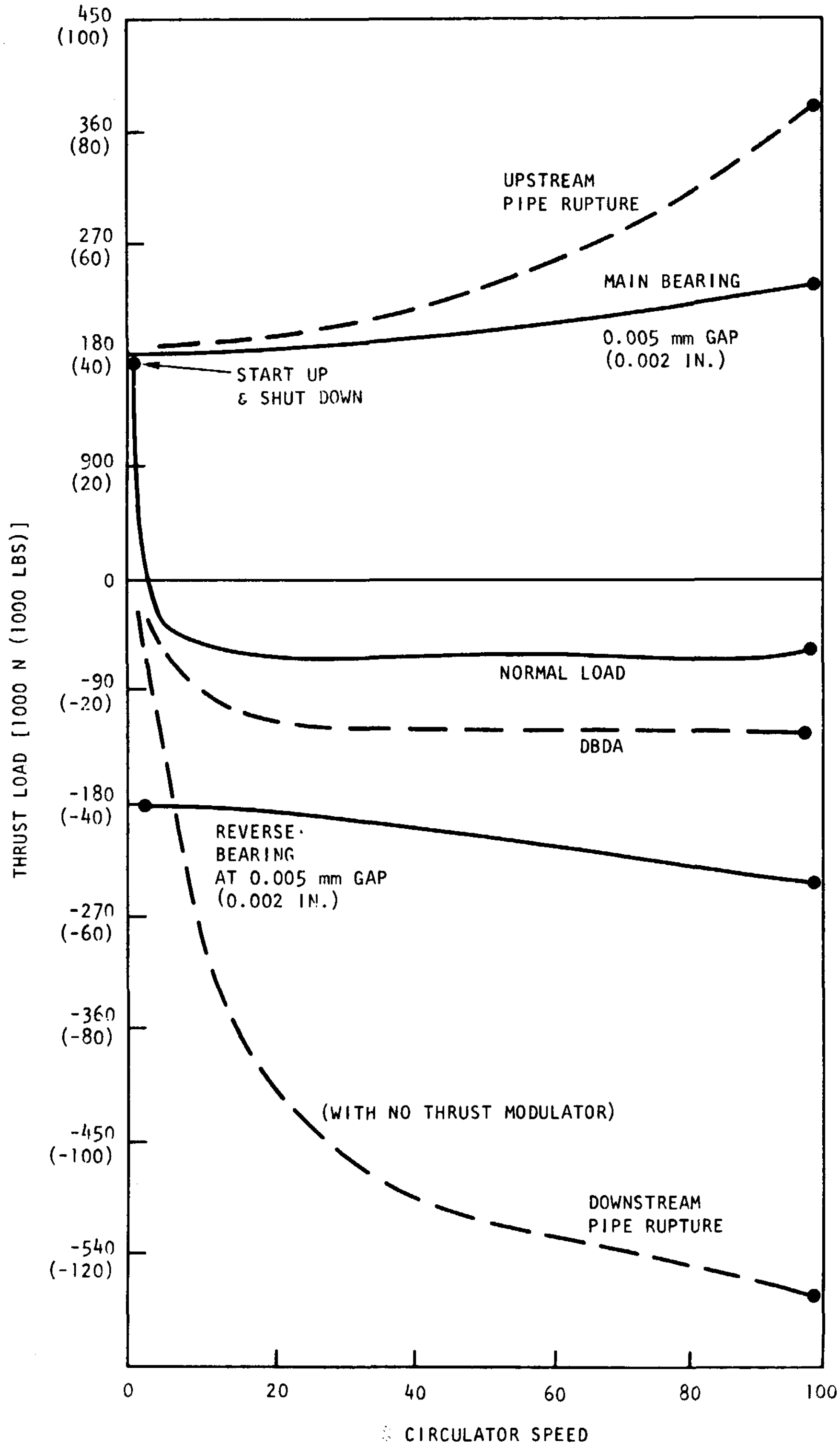


Fig. 24. Thrust loads for various operating conditions without a thrust balancing system

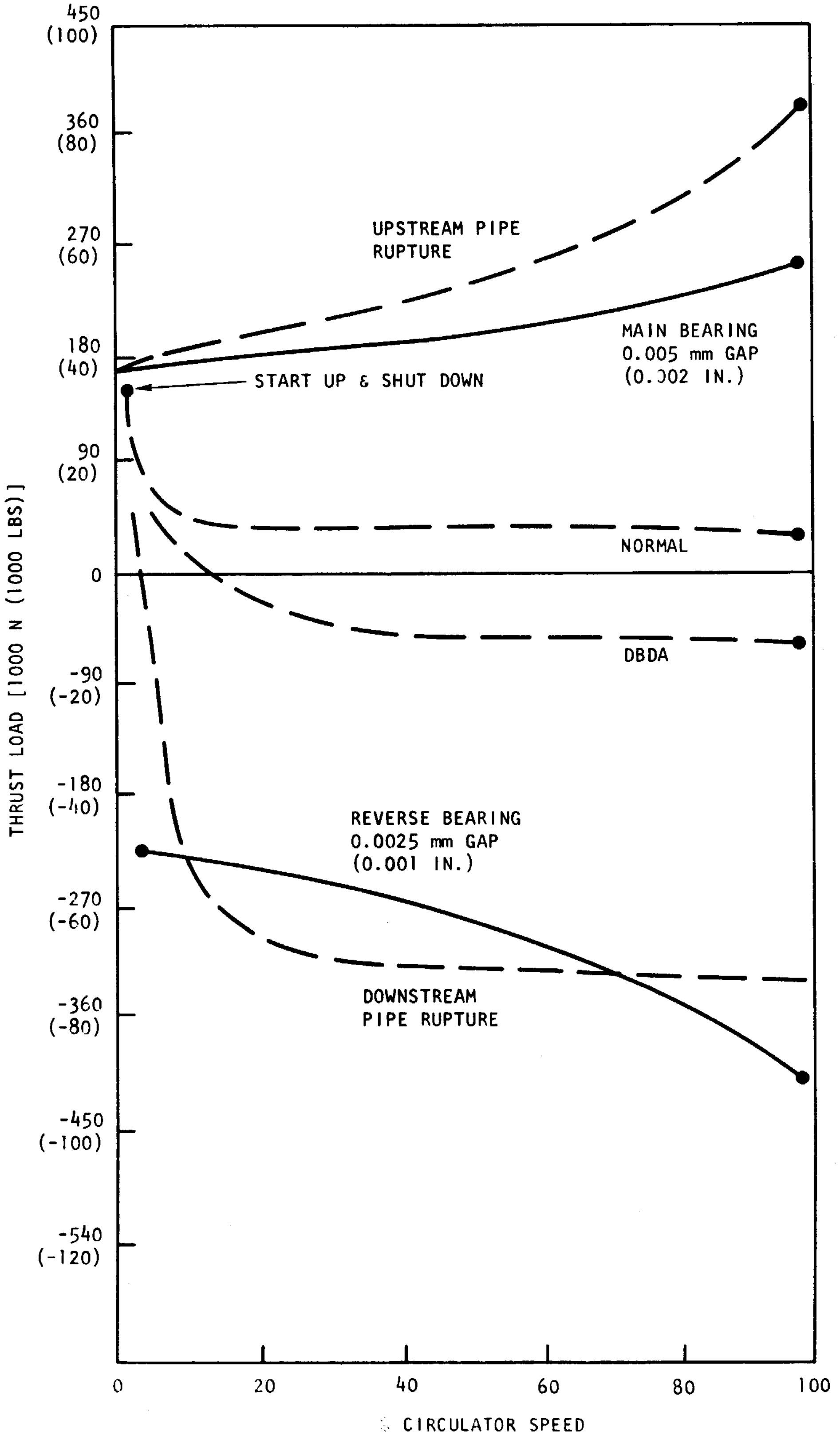
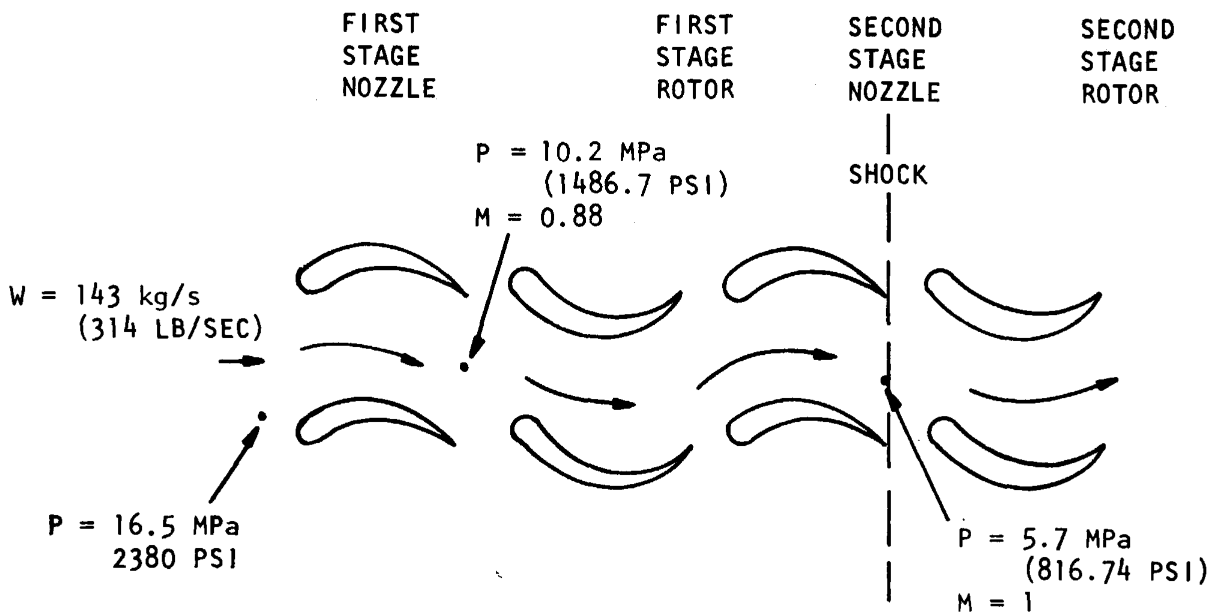


Fig. 25. Variable thrust loads with thrust balancing system



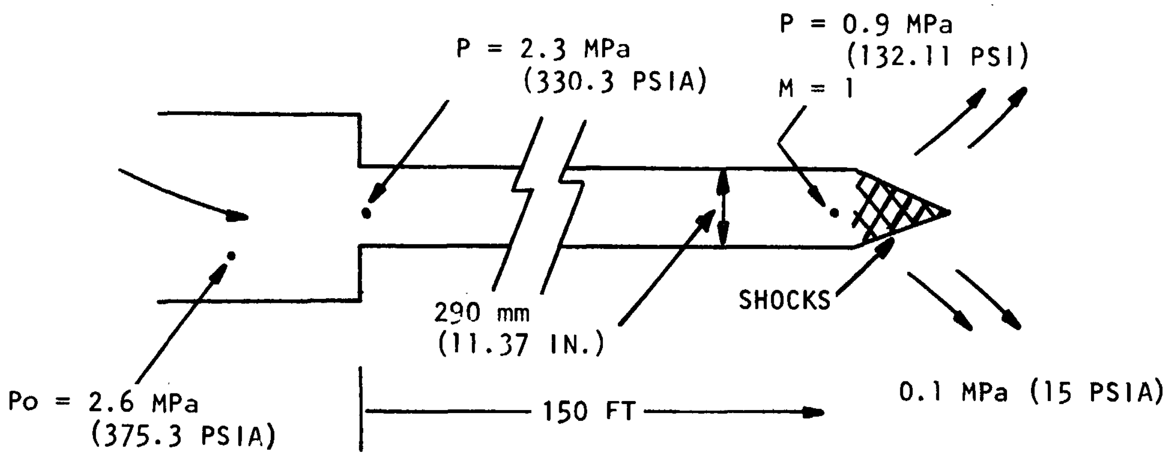


Fig. 26. Pressure distribution through turbine for downstream steam pipe rupture

turbine second stage will be about 486 psi (3.35 MPa). The pressure at the breaking point will be 132 psi (0.9 MPa); minimum back pressure at the turbine will be 330 psi (2.28 MPa).

Isentropic flow equations were used to determine the resulting flow condition; the turbine blading was assumed to be pure impulse and 100% efficient in adsorbing the flow energy. It was also assumed that all nozzle velocity is converted to work.

Assume 1st stage goes sonic then this flow must pass through the second stage nozzle at the sonic or less:

$$\frac{\rho_*}{\rho_o} = \left(\frac{2}{k+1}\right)^{\frac{1}{k+1}} = 1.597$$

or the area ratio from the second stage must be greater than 1.597. The actual area ratio is 1.498. Thus sonic flow from the second stage limits the mass flow. The approximate flow situation is shown in Figure 27.

A break point common to all three circulators would be 150 feet (46 m) down stream. A 12 inch (300 mm) pipe schedule 80 would give a flow velocity of 200 ft per second (61 m/sec) for normal conditions. For these conditions the flow is limited by the sunk conditions at the exit of the second stage turbine nozzles.

For a given long straight duct fed by a converging nozzle there is a maximum flow and minimum inlet pressure for a given reservoir pressure no matter how low the back pressure on the system.

The flow situation is illustrated in Figure 28.

For the down stream pipe rupture all required information is known or can be assumed to provide a solution. A graphic solution is shown in Figure 29.

The calculated turbine back pressure is 330 psia (2.3 MPa); the pressure at the breaking point is 132 psia (.9 MPa); resulting circulator overspeed 132%.

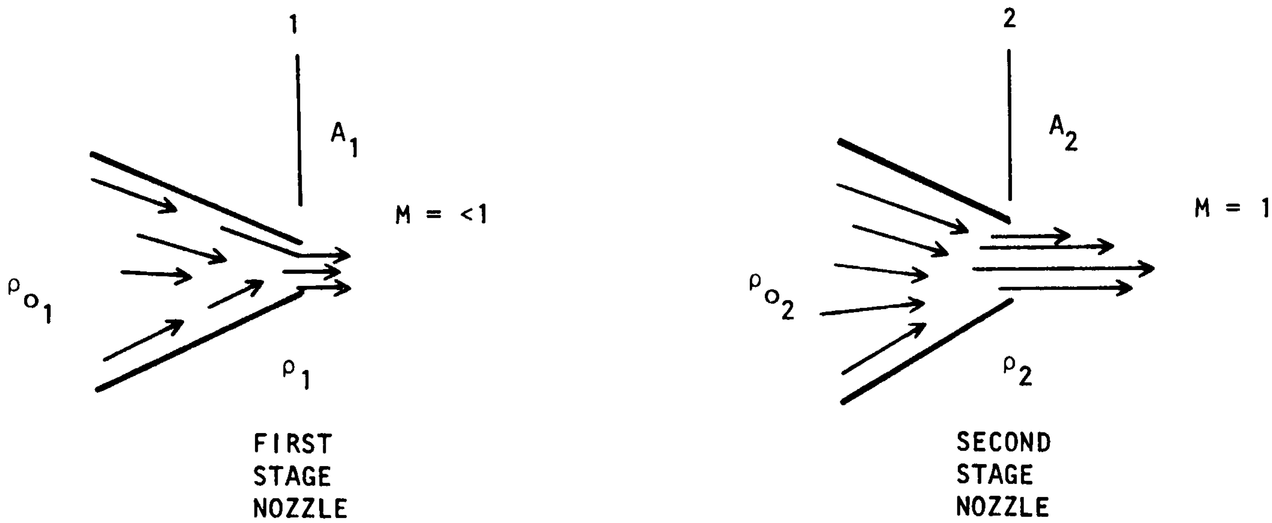


Fig. 27. Nozzle flow conditions

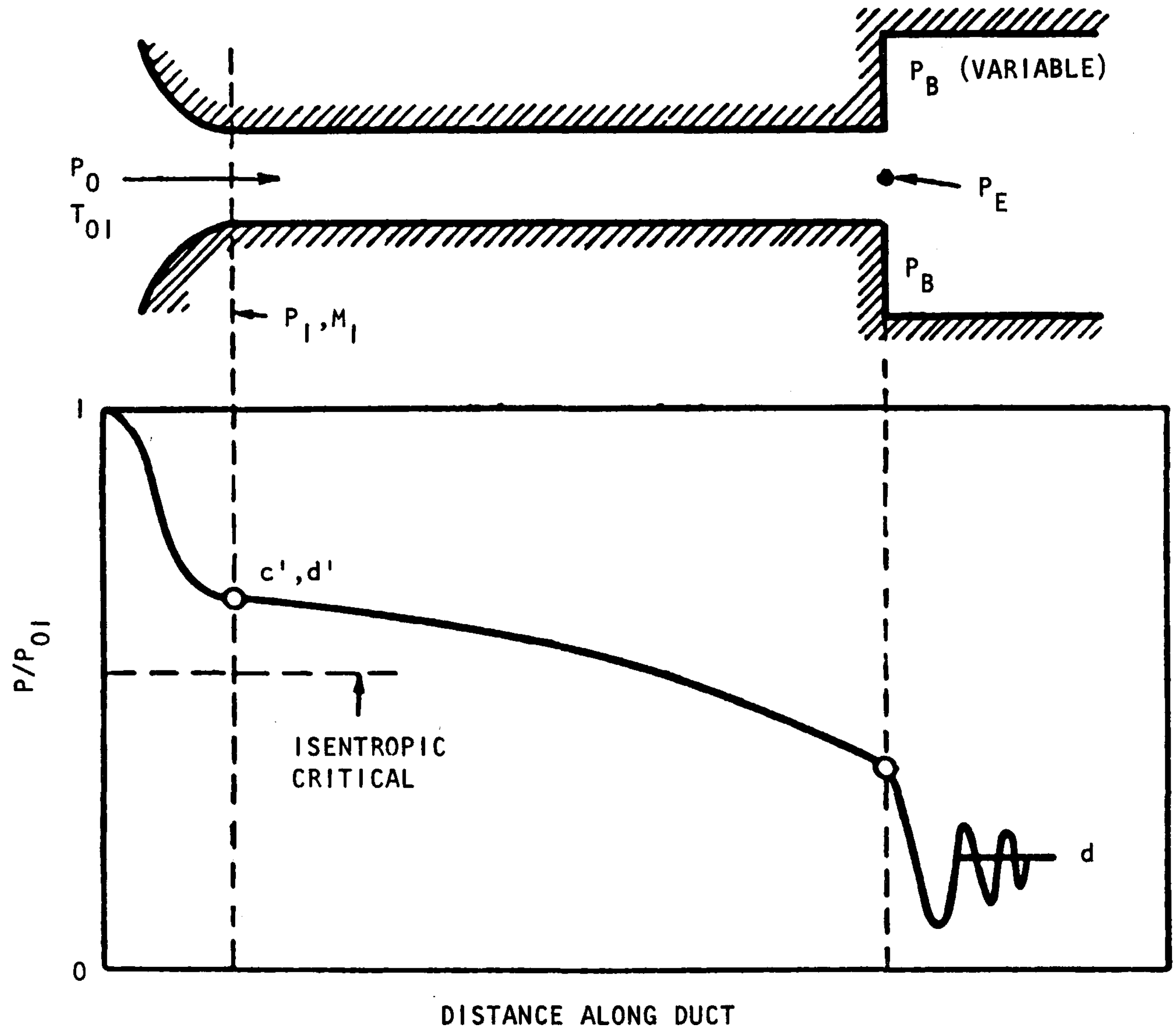
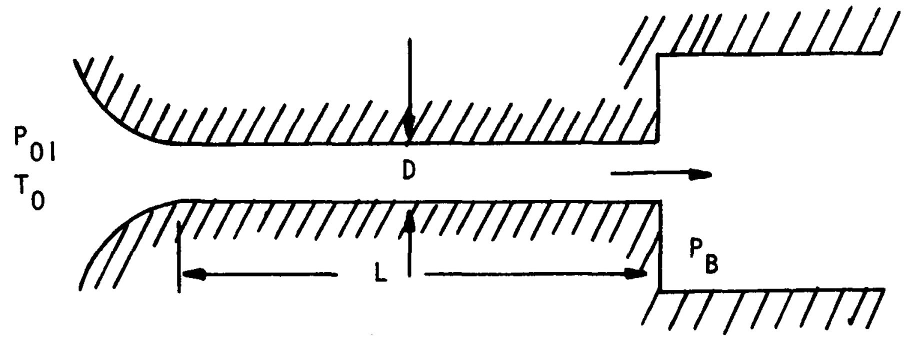


Fig. 28. Performance of duct fed by converging nozzle



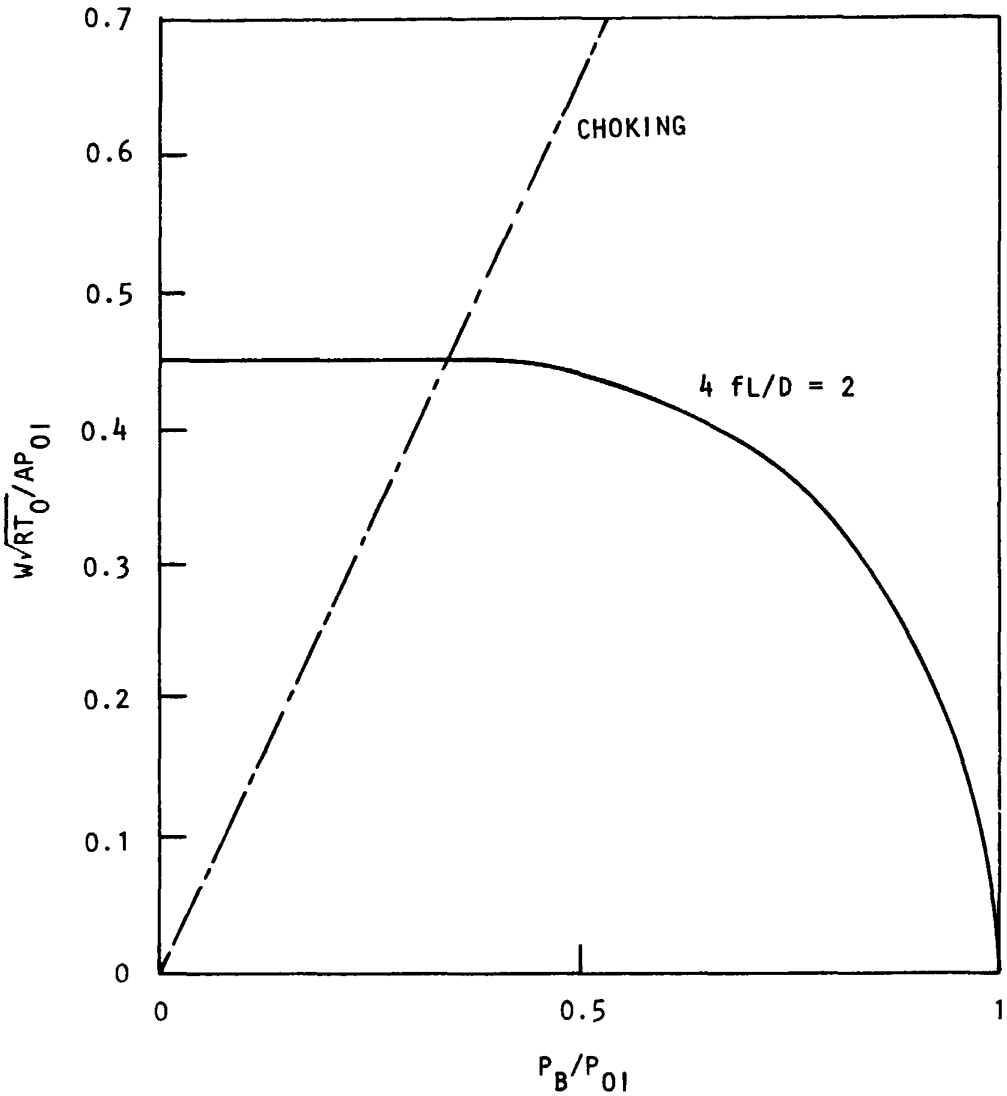


Fig. 29. Dimensionless flow parameter versus ratio of exhaust pressure to supply pressure

Circulator Thrust Loading Resulting from Down Stream Steam Pipe Rupture

The steam pipe rupture described previously will result in a worst case increase in the thrust load of about 180,000 lbs (800,000 N). This load can be reduced to a net thrust of 75,000 lbs (333,000 N) through the use of a thrust balancing device. The proposed scheme is shown in Figure 30. The resulting thrust loads for various operating conditions are:

	Thrust - 1000 lbs. (1000 N)					
					Net	
Condition	Turbine	Compressor	Shaft	Shaft & Balancer	With Balancer	Without Balancer
100% Speed Normal	-46(205)	+55(245)	-9(40)	-5(22)	+4(18)	0(0)
100% Speed Pipe Break	-226(1000)	+55(245)	-6(27)	+95(420)	-75(333)	-177(788)
133% Speed Pipe Break	-226(1000)	+89(396)	-6(27)	+95(420)	-41(182)	-143(635)

Maximum Upward Thrust

The upset and accident conditions that can result in a large upward thrust were investigated. The following three were found to be the most critical:

1. Scram from full load. Here, the worst differential between the helium and the steam pressure occurs after about 240 sec, when the steam inlet pressure is reduced to about 2150 psia (14.8 MPa) while the helium pressure is reduced to approximately 26 psia (.18 MPa). However, the steam pressure can be throttled in this condition to obtain small axial thrust loads.

2. Upstream steam pipe rupture. In this case the turbine pressure could rapidly decay to atmospheric pressure, with the helium pressure remaining at about 1250 psia (8.6 MPa).

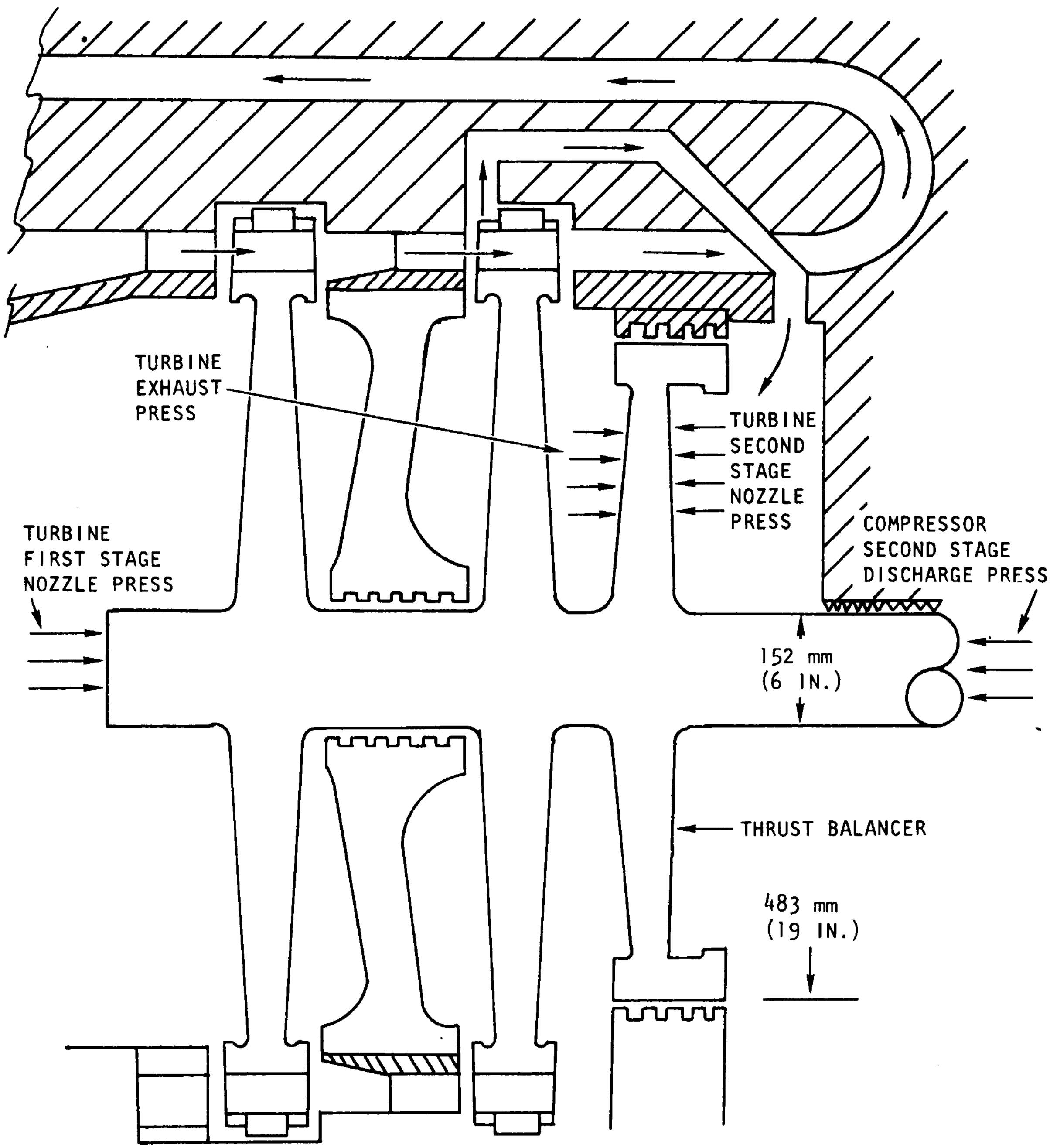


Fig. 30. Thrust balancer for downstream pipe rupture

3. Start-up (zero speed) and shutdown conditions. In this case full helium pressure is acting on the circulator end of the shaft with no or reduced steam pressure acting on the opposite end.

The ability to shut down the circulator without damage under all of the above conditions must be guaranteed. Case 2 sets the design condition for the thrust bearing. Therefore, the worst upward shutdown thrust is about 33,000 lbs. (147,000 N).

Maximum Downward Thrust

The only conditions that can lead to sizable downward thrust are a fast primary loop depressurization (27,000 lbs) and a down stream steam pipe rupture (75,000 lbs). In all other conditions the steam pressure decreases faster than the reactor pressure and no downward thrust can occur.

Following a fast primary loop depressurization, the reactor is scrammed either by low primary pressure or by high containment pressure. The latter case would lead to the highest downward thrust and is therefore considered here. Although no firm information is presently available regarding the scram set point, it is expected that scram will be initiated before the containment pressure becomes 10 psi (.07 MPa) higher than normal. This will occur within 20 sec or less. By superimposing the DBA depressurization curve (Figure 31) on a scram reheat pressure transient (with a 20 sec scram lag), it is found that the highest pressure differential between steam and helium occurs 240 sec after scram initiation. At this moment the available steam pressure is about 1700 psia (11.8 MPa), while the reactor pressure is about 25 psia (.17 MPa). However, there is sufficient time to throttle the steam pressure to reduce the thrust loading. The net downward thrust load was calculated to be less than 27,000 lbs (120,000 N) for DBDA operating conditions. However, the downstream steam pipe rupture accident described previously is worse than the DBDA load conditions with a 75,000 lbs downward thrust load. This latter load could be reduced by changing the diameter of the thrust modulator.

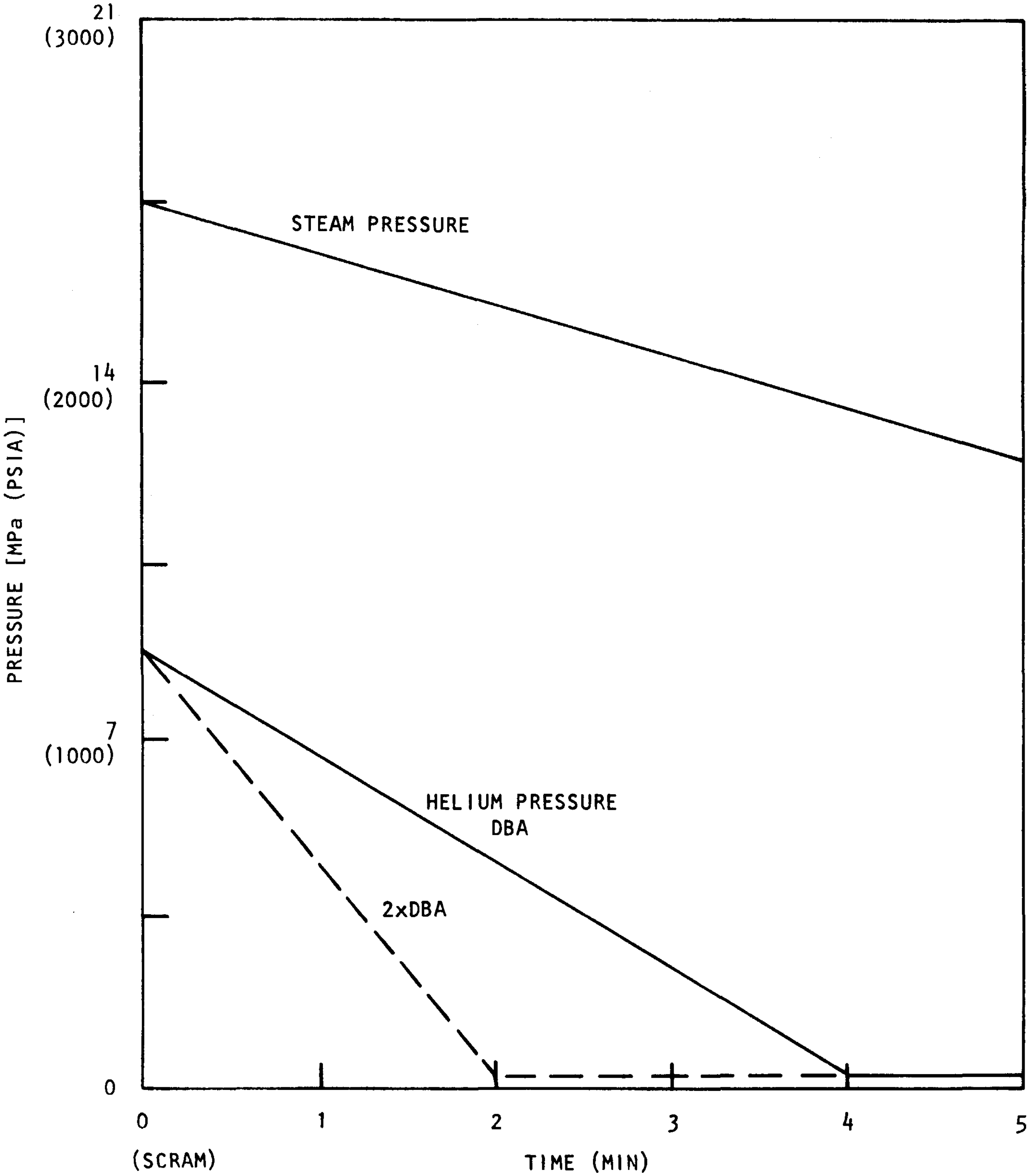


Fig. 31. Circulator and steam pressure transients following a fast depressurization accident (DBA)

5.3 Overspeed and Critical Speed Conditions

As described below, a number of overspeed conditions are possible. However, in determining critical design speeds only single failure conditions will be considered.

Some investigation into the potential requirements for main circulator overspeed for core cooling with a DBDA reveal that:

> To meet the presently stated core cooling requirements, two main circulators must operate at about 130% speed at 200 seconds following the initiation of the accident. This will require about 700 compressor shaft horsepower at the depressurized conditions existing at that time.
>
> The effect on component stress levels as a result of circulator overspeeds during accident conditions, including DBDA, is to be investigated when more detailed overspeed requirements become available.
>
> A downstream steam pipe rupture accident for the series flow turbine drive was investigated to determine the momentary effect on steam flow, circulator speed and thrust loads. It was concluded that under the worst circumstances the exit pressure acting across the turbine disk is about 400 psia (2.76 MPa) resulting in overall thrust loads which would be unacceptable at the thrust bearing. The momentary overspeed during such an accident would be 133% of normal operating speed, Figure 32.

Both the effects on the remaining operating circulators from a single circulator steam pipe failure and from a common main steam pipe rupture were investigated. It was concluded that by using a thrust modulator which will counteract the effects of even a total loss of steam back pressure to one circulator it will be possible to maintain reasonable thrust loads during all steam pipe rupture accidents

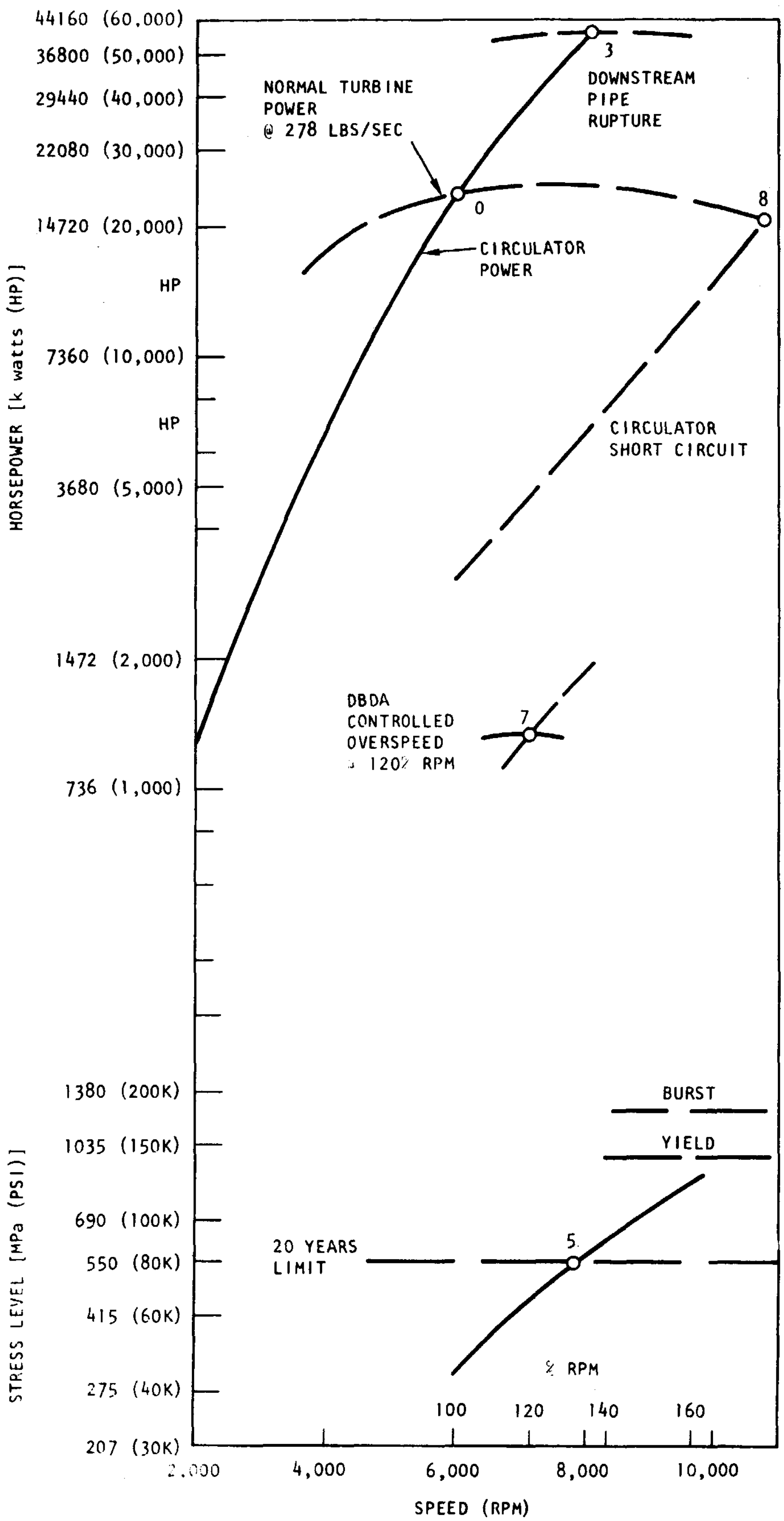


Fig. 32. Circulator and turbine power and stress conditions versus speed

and still maintain low thrust loads during normal operation. The thrust modulator is automatically balancing the net thrust loads by two different steam pressure levels each connected to the two sides of a rotating disk on the circulator shaft and separated by a seal arrangement. The two steam pressure levels are obtained by an open connection to two different sections of the steam flow passage through the turbine stages.

A sudden reduction of the required pressure increase of the circulator would result in higher flow rates and rotational speed. The worst condition would be a rupture between the high and low helium pressure sides near the circulator. The circulator speed would increase until the combined circulator and bearing horsepower equaled the turbine horsepower at about 10,800 rpm (see Figure 26). Note that the turbine horsepower is almost constant with increased rotational speed, because during normal operation the turbine horsepower is limited by the controlled steam flow, regardless of speed.

For a DBDA condition, with the helium pressure equal to the containment pressure, the rotational speed could increase to 7,200 rpm, as shown in Figure 26. Assuming the turbine pressure is limited to 290 psi (2.0 MPa).

The above examples indicate that high overspeeds, with subsequent high stress levels, are possible in the event of various system failures. A number of other critical operating conditions, given in Table 8, must also be considered. Some of the listed conditions are academic, since only a single failure condition will be considered. A redundant control system would eliminate the possibility of any excessive overspeed condition. The possible overspeed conditions listed in Table 8 are briefly described below.

The setting of n_1 is an arbitrary reasonable margin to avoid frequent trips. n_2 is based on an arbitrary condition above the overspeed trip point to protect against material failures during normal operation up to the overspeed trip point. n_3 is the maximum speed of the turbine based on choked flow in the nozzle in the case of loss of downstream pressure caused by a pipe rupture. n_4 is based on the minimum limiting yield strength of the blade or disk material. n_5 is an arbitrary failure speed at which unexpected failures might occur due to long life, even though all stresses are calculated to reach their limiting values at much higher speeds. n_6 is the runaway speed in the case of a total loss of helium coolant pressure (DBDA with containment backpressure). n_7 is the maximum speed during a DBDA with the steam pressure limited to 290 psi (2.0 MPa). n_8 is the overspeed which is caused by a wall rupture between the high and low helium pressure sides near the circulator. Finally n_9 is the first order critical speed, which is the speed at which the rotating speed coincides with the first natural oscillating frequency of the rotor shaft.

The above ratios indicate that n_4, n_6, n_7, n_8 and n_9 are far above any speed limitations of the control system and are not subject to testing. n_3, in the case of the GCFR, is within safe limits even if the control overspeed protection system were to fail and is therefore not critical. n_0 and n_1 are controlled speeds. In the case of the GCFR circulator, n_1, n_3 and n_5 are the only speeds for which the circulator must be tested. n_5 is a limiting cycle condition of, for example, 100 cycles during which the circulator is brought up to the n_5 speed and immediately reduced to operating speed. Since n_5 represents a worse condition than n_2 or n_3, no test is necessary for an n_2 or n_3 operating condition.

A design overspeed of 130% rpm would allow a DBDA condition using the main circulators for cooling. A downstream steam pipe rupture would cause an overspeed of 133% rpm. The turbine power is limited by the choked flow conditions. During any other accident condition the control system limits the overspeed to 110% rpm.

Based on the data presented in Table 8, it can be concluded that the circulator will operate within safe limits during its expected life span provided only single failures would occur during any given incident and that a redundant control system would always be in full operation. However, it should be noted that the overspeed conditions listed would normally not cause a machine rupture failure. The yield limit would be exceeded in the worst case, but the stress levels would not exceed rupture levels.

TABLE 8

POSSIBLE OVERSPEED CONDITION

	N/N_o
n_0 = design speed	1.00
n_1 = overspeed protection (trip)	1.10
n_2 = disk catcher speed	1.50
n_3 = downstream steam pipe rupture	1.33
n_4 = machine design failure speed	1.70
n_5 = expected failure speed after 20 years	1.40
n_6 = unload runaway speed (loss of helium pressure with steam system intact)	2.30
n_7 = DBDA with controlled steam pressure and speed	1.20
n_8 = loss of back pressure	1.80
n_9 = first order critical speed	$>$ 1.40

6. CONCLUSIONS

Based on the study described herein, it is concluded that it is possible to design a two stage main circulator, using stainless steel blades, which will satisfy the requirements for circulating helium at a high efficiency of nearly 80%. The steam turbine is somewhat more difficult to design because it is restricted by the operating characteristics of the circulator. However, it is possible to design a steam turbine with an 80% efficiency and practical dimensions and operating characteristics. The chosen design parameters for the circulator and the turbine are listed in Table 1. The turbine parameters were chosen to obtain high efficiency, acceptable rotor blade camber (less than six), a turbine blade height in excess of 13 mm (0.5 in.), near-zero reaction at the hub, and a nozzle and blade angle within the practical limits stated above. The upstream axial velocity and the diameter were fixed by these limitations. Turbine efficiencies higher than 80% would result in too-short turbine blades and a larger wheel diameter. Efficiencies below 80% would result in lower turbine exit pressures and undesirable blade cambers.

The flow ducts, diffusers, bearings, and rotating disks are similar in design to previously designed General Atomic systems. No problems were encountered in these systems. The maximum upward shaft thrust will be 33,000 lb (147,000 N) and the worst downward thrust will be 10,000 lb (44,500 N). It is not possible to reduce these loads by pressure equalization systems across the circulator and turbine disks. Also, it is necessary to introduce the high-pressure steam in the inside pipe of the two concentric pipes leading into and out of the circulator turbine. Otherwise, the maximum load conditions will be substantially higher than the values stated above.

7. NOMENCLATURE

a = aspect ratio or h/c

A = cross-sectional flow area, ft^2 (cm^2)

c = blade camber

c = speed of sound, ft/sec (mm/sec)

c = blade chord, in. (mm)

d = diameter, ft. (mm)

d = diffusion factor

D = diameter, ft. (mm)

F = force, lb (kg)

h = blade height, in. (mm)

M = Mach number

N = rotational speed, rpm

p = pressure, psi (MPa)

Δp = pressure increase, psi (MPa)

P = force, lb (N), or power, watts

PWL = sound power level, dB

SPL = sound pressure level, dB

S = blade solidity

U = blade speed, ft/sec (mm/sec)

V = velocity, ft/sec (mm/sec)

W = flow rate, lb/sec (kg/sec)

W_s = shaft weight, lb (kg)

α = angle, degrees

γ = gas density, lb/ft^3 (kg/mm^3)

δ = pressure ratio

η = efficiency

θ = temperature ratio

ρ = density, lb/ft^3 (kg/mm^3)

τ = blade thrust, lb (N)

ω = angular speed, rad/sec

Subscript C = circulator compressor

Subscript T = turbine

σ_{freq} = blade stress level which results in resonance frequencies above imposed frequencies

8. BIBLIOGRAPHY

"Aerodynamic Design of Axial-Flow Compressors," National Advisory Committee for Aeronautics Report NACA RM E56B03a, Vol. II, Chap. VII, August 1956.

Ainley, D. C., and G. C. R. Mathieson, "A Method of Performance Estimation for Axial Flow Turbine," Aeronautical Research Council Report R & M - 2074, 1957.

Cavallaro, L., "On the Stress Analysis of an Axisymmetrical Disk with Variable Thickness," USAEC Report GA-6267, General Dynamics, General Atomic Division, September 20, 1965.

Dunavant, J. C., and J. R. Erwin, "Investigation of a Related Series of Turbine-Blade Profiles in Cascade," National Advisory Committee for Aeronautics Report NACA TN 3802, October 1956.

Felix, A. R., "Summary of 65-Series Compressor-Blade Low-Speed Cascade Data by Use of the Carpet-Plotting Technique," National Advisory Committee for Aeronautics Report NACA TN 3919, February 1957.

Lieblein, S., "Loss and Stall Analysis of Compressor Cascades," J. Basic Eng. 81, 387-400 (September 1959).

Lieblein, S., F. C. Schwenk, and R. L. Broderick, "Diffusion Factor for Estimating Losses and Limiting Blade Loadings on Axial Flow Blade Compressor Elements," National Advisory Committee for Aeronautics Report NACA RM E53D01, 1953.

Miser, J. W., W. L. Stewart, and W. J. Whitney, "Analysis of Turbomachine Viscous Losses Affected by Changes in Blade Geometry," National Advisory Committee for Aeronautics Report NACA RM E56F21, October 1956.

Montakhab, A., and R. M. Sullivan, "DOVTAL, A Computer Program for the Stress Analysis of Dovetail-Type Blade-Roots and Disk Rim Grooves in Axial Flow Turbomachines," Gulf General Atomic, unpublished data.

Stewart, W. L., and D. G. Evans, "Analytical Study of Losses at Off-Design Conditions for a Fixed Geometry Turbine," National Advisory Committee for Aeronautics Report NACA RM E53K06, 1953.

Stewart, W. L., W. J. Whitney, and R. R. Wong, "A Study of Boundary Layer Characteristics of Turbomachine Blade Rows and Their Relation to Overall Blade Loss," J. Basic Eng. 82, 588-592 (September 1960).

Yampolsky, J., "Investigation of Turbocirculator for TARGET, A 1000-MW(e) Reactor (Methodology)," General Dynamics, General Atomic Division, unpublished data.

Sweifel, O., "Optimum Blade Pitch for Turbomachine with Special Reference to Blades of Great Curvature," The Engineers Digest 7, 358-360 (November 1946); 381-383 (December 1946).

Mouritzen, G., "Preliminary Design of a Helium Circulator and Steam Turbine for a 300 MW (e) GCFR Demonstration Plant," General Atomic Company, unpublished data.

Mouritzen, G., "Preliminary Design of a Helium Circulator and Steam Turbine for a Typical 750/1500 MW(e) Commercial Type GCFR Plant," General Atomic Company, Report No. GA-A13504, December 1975.