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HELIUM COMPRESSOR AERODYNAMIC DESIGN CONSIDERATIONS FOR MHTGR CIRCULATORS

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Abstract

Compressor aerodynamic design considerations for both the main and shutdown cooling circulators in the Modular High-Temperature Gas-Cooled Reactor (MHTGR) plant are addressed in this paper. A major selection topic relates to the impeller type (i.e., axial or radial flow), and the aerothermal studies leading to the selection of optimum parameters are discussed. For the conceptual designs of the main and shutdown cooling circulators, compressor blading geometries were established and helium gas flow paths defined. Both circulators are conservative by industrial standards in terms of aerodynamic and structural loading, and the blade tip speeds are particularly modest. Performance characteristics are presented, and the designs embody margin to ensure that pressure-rise growth potential can be accommodated should the circuit resistance possibly increase as the plant design advances. The axial flow impeller for the main circulator is very similar to the Fort St. Vrain (FSV) helium compressor which performs well. A significant technology base exists for the MHTGR plant circulators, and this is highlighted in the paper.

1. INTRODUCTION

Details of the circulator configurations (Ref. 1) that are responsive to the system requirements (Ref. 2) have been addressed as separate topics, and this paper focuses on aerodynamic design considerations for the compressor impellers in the two circulators. Both axial and radial flow impeller compressors have been utilized in gas-cooled reactors (Refs. 3 to 6), and there are no firm ground rules that mandate the best type. The compressor cannot be designed in an isolated manner, but must be integrated in the reactor system to satisfy requirements in the areas of thermal-hydraulics, gas-flow path compatibility, drive type, and possible future pressure-rise growth potential. Two circulators are addressed in this paper, and the considerably different requirements and criteria led to the selection of different impeller types; namely, an axial flow compressor for the main circulator and a radial flow configuration for the small shutdown cooling machine. The studies leading to the selection of the two compressor types are the focal point of this paper.

2. CIRCULATOR BACKGROUND

In the first generation of carbon dioxide-cooled plants (Magnox Reactors) built in the United Kingdom (Ref. 5) and France (Ref. 7), the modest levels of circuit pressure loss were compatible with the selection of a single-stage axial flow compressor. Various electric motor and steam turbine drives were used for these machines. With the introduction of the AGR commercial power plants, the higher resistance in the carbon dioxide circuits were beyond the capability of a single-stage axial flow compressor, and the choice lay between multistage axial or radial flow compressors. The simplicity and ruggedness of the radial impeller was a deciding factor in the selection of this type of machine (Ref. 8).

The selection of reactor coolant also has an effect on the circulator design, and in the case of the aforementioned units the very dense carbon dioxide yielded low values of adiabatic head (i.e., pressure rise/density). In the early helium-cooled reactors (Dragon, AVR, Peach Bottom 1), the circulator head rise favored the selection of radial flow compressors. In the case of the circulators for commercial pebble bed reactors, the very high circuit resistance associated with this type of reactor core essentially dictated the use of radial flow compressors (Ref. 9).

In the case of the FSV plant and the units developed for Delmarva, the following factors led to the selection of an axial flow impeller: (1) low pressure rise characteristic of the prismatic core, (2) compatibility with high speed steam turbine drive, and (3) requirement to minimize the machine diameter for vertical installation within the prestressed concrete reactor vessel.

In reviewing the technology bases, it is clear that the radial flow compressor is dominant in the gas-cooled reactor field, and its simplicity is perhaps its greatest attribute. In the conceptual design of the MHTGR plant, there was no attempt to "force-fit" a given circulator configuration into the reactor circuit. The following sections highlight the results of studies to identify the most optimum type for the two circulators in the MHTGR plant based on an annular prismatic reactor core.

3. MAIN CIRCULATOR

3.1. INSTALLATION

As illustrated in Fig. 1, the main circulator is installed in the top head of the steam generator vessel. The main circulator facilitates transfer of reactor thermal energy to the steam generator. The system is capable of decay heat removal in pressurized and depressurized modes of operation.

3.2. REQUIREMENTS

Comprehensive requirements for the circulator have been discussed previously (Ref. 2), and boundary conditions necessary for the design of the circulator are given in Table 1.

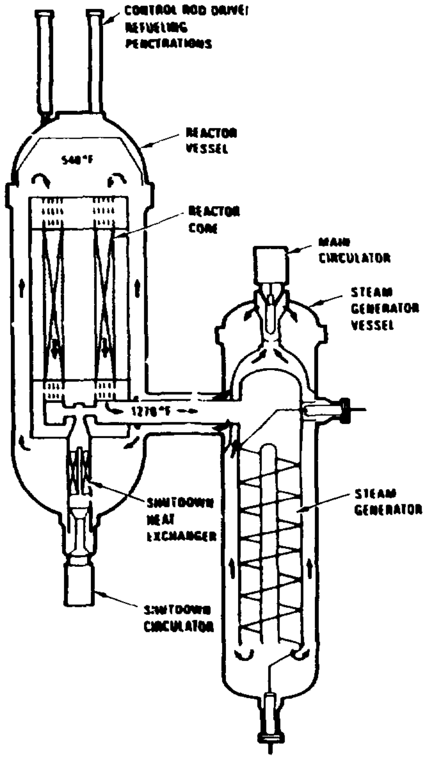


Fig. 1 Main Helium Circulator Installed in Heat Transport Loop

3.3. AXIAL VERSUS RADIAL FLOW COMPRESSOR AERODYNAMIC CONSIDERATIONS

A summary of the major factors in the selection of the impeller type is given in Table 2. Early in the comparative study, it was concluded that both types could meet the system requirements, the designs being based on demonstrated and proven technology.

3.3.1. Radial Flow Impeller

Initial studies of a pebble bed variant of the MHTGR confirmed the findings of similar work done in Germany that a radial flow compressor

TABLE 1. MAIN CIRCULATOR DESIGN REQUIREMENTS

- PROVIDE HELIUM FLOW OF 348 LB/SEC (158 Kg/SEC)
- PROVIDE HELIUM PRESSURE RISE OF 13.2 PSI (91 kPa)
- PROVIDE CAPABILITY TO CIRCULATE HELIUM FOR PRESSURIZED AND DEPRESSURIZED SYSTEM CONDITIONS
- FLOW CONTROL BY VARIABLE SPEED MACHINE
- ENSURE STABLE OPERATION OVER WIDE SPEED RANGE (5%–110%)
- INCORPORATE PRESSURE RISE GROWTH CAPABILITY
- INSTALL CIRCULATOR IN LOW TEMPERATURE PART OF CIRCUIT
- BASE DESIGN ON ESTABLISHED AND PROVEN TECHNOLOGY
- ESTABLISH MACHINE ENVELOPE TO FACILITATE REMOVAL AND REPLACEMENT IN SPACE ABOVE STEAM GENERATOR VESSEL
- STRESS LEVELS TO BE COMMENSURATE WITH 40 YEAR LIFE REQUIREMENT

was necessary to accommodate the high system pressure loss. Details of the radial flow machine are given in Fig. 2.

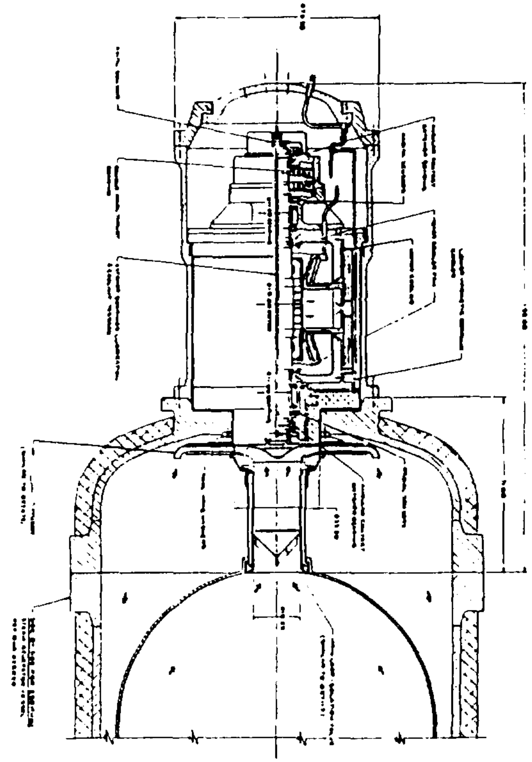
In performing scoping studies on compressors, a particularly useful relationship involves the portrayal of data on a specific speed-diameter array, the approach having been developed by Balje (Ref. 10). An example of this relationship for radial compressors is given on Fig. 3, and the obvious goal is to establish aerodynamic parameters to give a design solution in the regime of high efficiency. Data points for operational machines are superimposed on the island array. In many cases, there are physical limitations (e.g., driver speed, envelope constraints, etc.) that do not permit operation in the island of maximum efficiency; nevertheless, the compressor performs well. The array should be regarded as a first step towards indicating whether a radial, mixed, or axial flow machine should be selected.

As an initial point (in the center of the efficiency island), a specific speed and diameter of 100 and 1.5 (in the units stated), respectively, would yield an impeller diameter of 1676 mm (66 in.) and a

TABLE 2. AXIAL OR RADIAL COMPRESSOR FOR HELIUM CIRCULATOR

CONSIDERATION	AXIAL FLOW COMPRESSOR	RADIAL FLOW COMPRESSOR
DESIGN STATUS	CONCEPTUAL DESIGN	CONCEPT EVALUATED
MEETING DESIGN PRESSURE RISE REQUIREMENT (13.2 PSI)	SINGLE STAGE ADEQUATE	SINGLE STAGE
PRESSURE RISE MARGIN CAPABILITY (25% ABOVE DESIGN)	CAN BE ACCOMPLISHED WITH SINGLE STAGE	SINGLE STAGE HAS CAPABILITY FOR MUCH HIGHER PRESSURE RISE (WELL SUITED TO PBR PLANT)
EXPERIENCE/TECHNOLOGY	COMPRESSOR VERY SIMILAR TO FSV IMPELLER THAT PERFORMS WELL	INDUSTRIAL COMPRESSORS
PERFORMANCE	SELECTED AXIAL COMPRESSOR NEAR OPTIMUM PARAMETERS	LARGE DIAMETER IMPELLER, AND LOW SPEED FOR EFFICIENT OPERATION
PRIMARY SYSTEM GAS FLOW PATH COMPATIBILITY	SELECTED FLOW DIRECTION THROUGH IMPELLER GREATLY REDUCES THRUST BEARING LOAD	GAS FLOW PATH RESULTS IN THRUST AND ROTOR WEIGHT BEING ADDITIVE
MACHINE ENVELOPE/WEIGHT	COMPACT SMALL DIAMETER MACHINE ASSEMBLY	INCREASED DIAMETER (WEIGHT ASSEMBLY)
SIMPLICITY/RUGGEDNESS	DEMONSTRATED IN FSV	SIMPLE ROTOR, VERY RUGGED
DESIGN CONSERVATISM	MACHINE WELL WITHIN STRESS LIMITS	CONSERVATIVE (LOW SPEED)
MEETS SYSTEM REQUIREMENTS	YES	YES
GAS-COOLED REACTOR EXPERIENCE	FSV EARLY MAGNOX PLANTS (CO_2)	PEACH BOTTOM, AVR, THTR 116 MACHINES BUILT FOR AGR (CO_2)
OVERALL SUMMARY	COMPATIBLE WITH REACTOR SYSTEM COMPRESSOR SIMILAR TO FSV UNIT THAT PERFORMS WELL NEAR OPTIMUM	RADIAL COMPRESSOR WELL SUITED TO PLANTS (SUCH AS THE PEBBLE BED REACTOR) WITH HIGH CIRCUIT PRESSURE LOSS

rotational speed of 2000 rpm. Bearing in mind the requirement for a compact machine assembly to facilitate ease of removal and replacement (Ref. 1), such a compressor would not be practical from the installation standpoint. A parametric array showing the impact of impeller diameter and speed is given on Fig. 4. Considering envelope constraints, an impeller diameter of 1118 mm (44 in.) with a rotational speed of 3600 rpm was tentatively selected. With a specific speed and diameter of 180 and 1.0, respectively (see Table 3 and Fig. 3), such a radial flow machine is considered viable, but as will be discussed below, it is physically larger than a comparable axial flow machine.



- MAIN CIRCULATOR FOR 250 MW(t) MODULAR PBR PLANT CONCEPT
  - W = 252 LB/SEC (114 Kg/SEC)
  - T_{in} = 491°F (255°C)
  - P_{out} = 1050 psi (7.24 MPa)
  - Δp = 33 psi (228 KPa)
- RADIAL IMPELLER
  - DIAMETER, 33.2 IN. (843 mm)
  - SPEED, 5240 RPM
  - POWER, 6800 HP [5073 kW(e)]

Fig. 2 Initial Radial Flow Circulator for 250MWt Pebble Bed Reactor

3.3.2. Axial Flow Impeller

From the onset of the study, it was quite clear that the compressor requirements were similar to those for Fort St. Vrain (Table 3). While the FSV circulator has experienced operating problems (Ref. 11), the impeller performs well, and there was an obvious incentive to take advantage of this. Since helium flow control is accomplished by variable speed (as opposed to variable inlet guide vane geometry), there is no obvious incentive to have a stator ahead of the rotor; in fact, it is

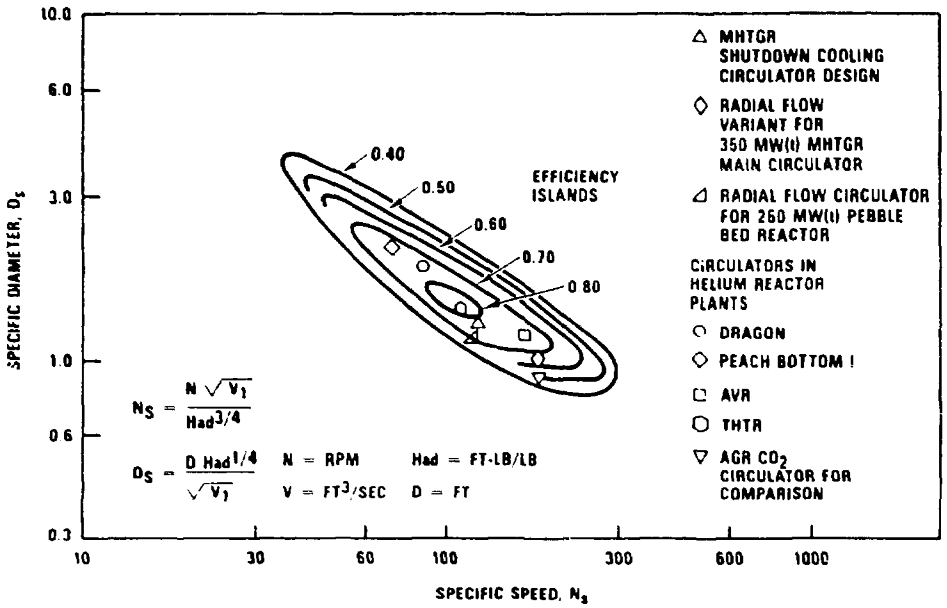


Fig. 3 Specific Speed-Diameter Array for Radial Flow Compressor

desirable to minimize the number of stages. As in the case of the FSV compressor, a rotor-before-stator configuration was selected, and the velocity diagram is shown on Fig. 5. In this zero inlet and exit swirl arrangement, the rotors impart the energy into the gas in the form of a velocity, and the trailing stators recover this energy by returning the gas velocity to the axial direction.

A preliminary aerodynamic design of the compressor was performed using a computer code developed by GA Technologies based on well-established design methods, such as those presented in NASA SP-36 (Ref. 12). The aerothermodynamic parameters for the axial flow machine are given on Table 4. A study was performed to address an obvious question, "If the system pressure loss was to increase (as a result of added

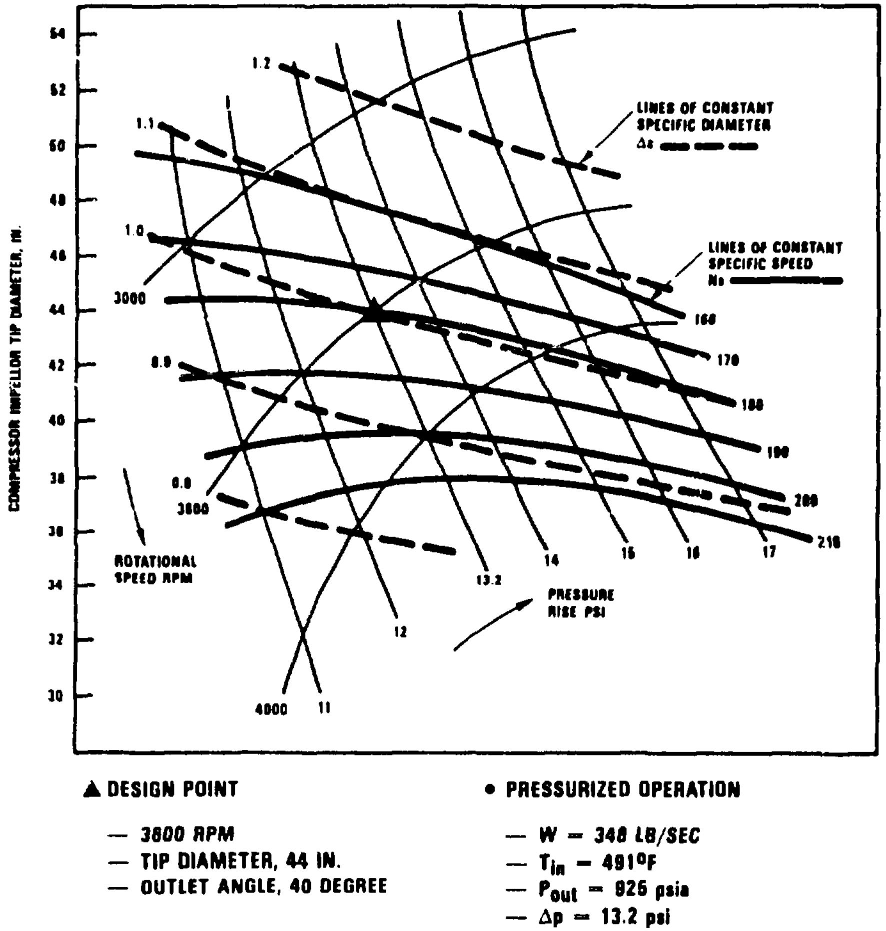


Fig. 4 Parametric Array for Main Circulator Radial Compressor

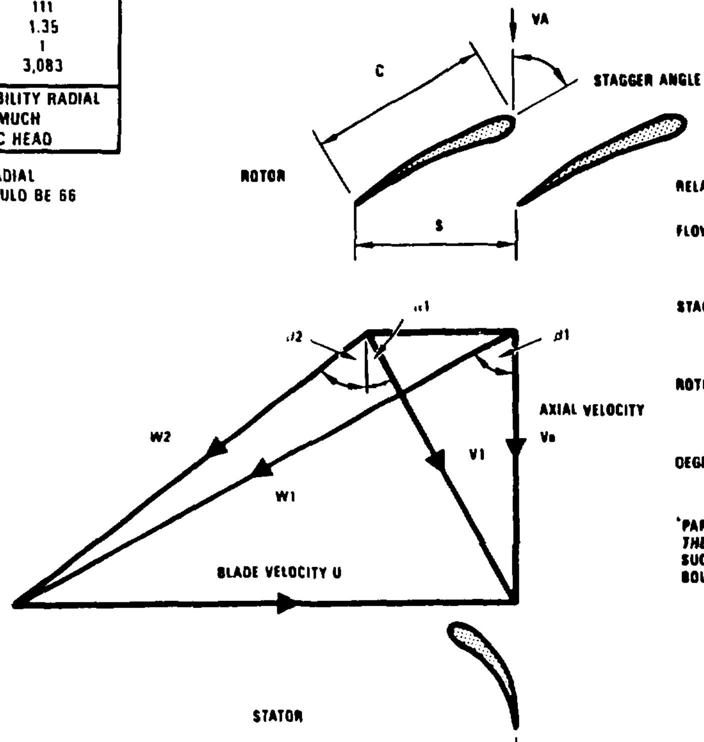
complexity as the plant design advances), could a single-stage axial compressor accommodate this?"

From the compressor map shown on Fig. 6, it was concluded that the proposed design had adequate surge margin, as was demonstrated in the

TABLE 3. COMPRESSOR COMPARISON FOR HELIUM CIRCULATOR

COMPRESSOR TYPE	AXIAL FLOW		RADIAL FLOW	
COMPRESSOR DESIGN	MHTGR MAIN CIRCULATOR	FSV CIRCULATOR	MHTGR MAIN CIRCULATOR	THTR CIRCULATOR
HELIUM FLOW, LB/SEC	348	278	DATA	112
INLET TEMP, °F	491	742	AS	482
INLET PRESSURE, PSIA	911.8	686	FOR AXIAL	551
PRESSURE RISE, PSI	13.2	14.0	FLOW	18.0
INLET DENSITY, LB/FT^3	0.358	0.213	COMPRESSOR	0.218
VOLUMETRIC FLOW, FT^3/SEC	972	1,304		514
ADIABATIC HEAD, FT	5,310	9,465		11,890
IMPELLER O/D, IN.	35.0	28.0	44.0[a]	35.4
SPEED, RPM	6,200	9,550	3,600	5,600
TIP SPEED, FT/SEC	947	1,167	891	865
SPECIFIC SPEED, N_s	311	360	180	111
SPECIFIC DIAMETER, D_s	0.80	0.64	1.0	1.35
NUMBER OF STAGES	1	1	1	1
POWER, HP	4,250	5,300	APPROX. 4,600	3,083
PRESSURE — RISE GROWTH CAPABILITY	● 25% Δp INCREASE CAN BE ACCOMMODATED WITH SINGLE AXIAL STAGE		● GOOD GROWTH CAPABILITY RADIAL STAGE CAN ACHIEVE MUCH INCREASED ADIABATIC HEAD	

[a]VALUE SELECTED FOR CONCEPTUAL DESIGN, RECOGNIZING ENVELOPE CONSTRAINT. FOR OPTIMAL RADIAL COMPRESSOR PRAMETERS (i.e., N_s = 100, D_s = 1.5) THE COMPRESSOR DIAMETER AND SPEED WOULD BE 66 IN. AND 2000 RPM, RESPECTIVELY, AND NOT PRACTICAL FROM INSTALLATION STANDPOINT.



RELATIONSHIPS FROM VELOCITY DIAGRAM

FLOW COEFFICIENT $\frac{Va}{U} = \frac{1}{\tan \beta 1} = \frac{1}{\tan \beta 2 + \tan \alpha 1}$

STAGE TEMPERATURE RISE $\frac{\Delta T}{U^2} = \frac{Va}{U} (\tan \beta 1 - \tan \beta 2) \frac{1}{g\ J\ Cp}$

ROTOR DIFFUSION FACTOR D* = $\left(1 - \frac{\cos \beta 1}{\cos \beta 2} + \frac{S}{C}\frac{\cos \beta 1}{2}\right.$ $(\tan \beta 1 - \tan \beta 2)$

DEGREE OF REACTION R = $\frac{Va}{2U} (\tan \beta 1 + \tan \beta 2)$

*PARAMETER RELATING TO AERODYNAMIC LOADING IN AN AXIAL FLOW COMPRESSOR. THE DIFFUSON FACTOR IS AN INDICATION OF BOUNDARY LAYER GROWTH ON THE SUCTION SIDE OF THE BLADE. WHEN D ≥ 0.60 THERE IS EVIDENCE OF BOUNDARY LAYER SEPARATION.

ROTOR BEFORE STATOR TYPE
(ZERO INLET AND EXIT SWIRL)

Fig. 5 Velocity Diagram for Axial Flow Compressor

TABLE 4. AEROTHERMODYNAMIC PARAMETERS FOR HELIUM COMPRESSOR

DESIGN DATA	PARAMETER
HELIUM FLOW RATE, KG/SEC (LB/SEC)	158 (348)
INLET TEMPERATURE, °C(°F)	255 (491)
INLET PRESSURE, MPa (PSIA)	6.29 (912)
CIRCULATOR PRESSURE RISE, KPa (PSID)	91 (13.2)
FLOW CONTROL	VARIABLE SPEED DRIVE
ADIABATIC HEAD, M (FT)	1618 (5309)
DIFFUSER EFFICIENCY, %	80
DIFFUSER AREA RATIO	0.24
OVERALL EFFICIENCY, % (TOTAL-TO-STATIC)	79.2
MOTOR SHAFT POWER, KWe(HP)	3170 (4250)
COMPRESSOR TIP DIAMETER, MM (IN.)	889 (35.0)
BLADE HEIGHT, MM (IN.)	88.9 (3.5)
HUB/TIP RATIO	0.80
TIP SPEED, M/SEC (FT/SEC)	289 (947)
AXIAL VELOCITY, M/SEC (FT/SEC)	125 (411)
ROTATIONAL SPEED, RPM	6200
FLOW COEFFICIENT, Va/Um	0.48
MEAN ROTOR SOLIDITY	1.02
ROTOR ASPECT RATIO	1.22
MEAN ROTOR MACH NUMBER	0.21
ROTOR DIFFUSION FACTOR (MAX)	0.42
DEGREE OF REACTION	0.85
TEMPERATURE RISE COEFFICIENT, $\Delta H/U_m^2$	0.29
SPECIFIC SPEED, Ns	311
SPECIFIC DIAMETER, Ds	0.80
HEAD COEFFICIENT, $H_{ad}/U^2/g$	0.19
DESIGN TECHNOLOGY	BASED ON PROVEN TECHNOLOGY

FSV machine. The results of the pressure rise growth potential are shown on Fig. 7. A parameter that is indicative of aerodynamic loading in an axial flow compressor is the diffusion factor, which is indicative of the extent of boundary layer growth on the suction side of the blade. In comparing the NASA design method with that of Howell (Ref. 13), it can be seen from Fig. 8 that the diffusion factor is a strong function of gas deflection (for a given solidity). There is evidence of boundary layer separation on the suction surface of the airfoil if the diffusion

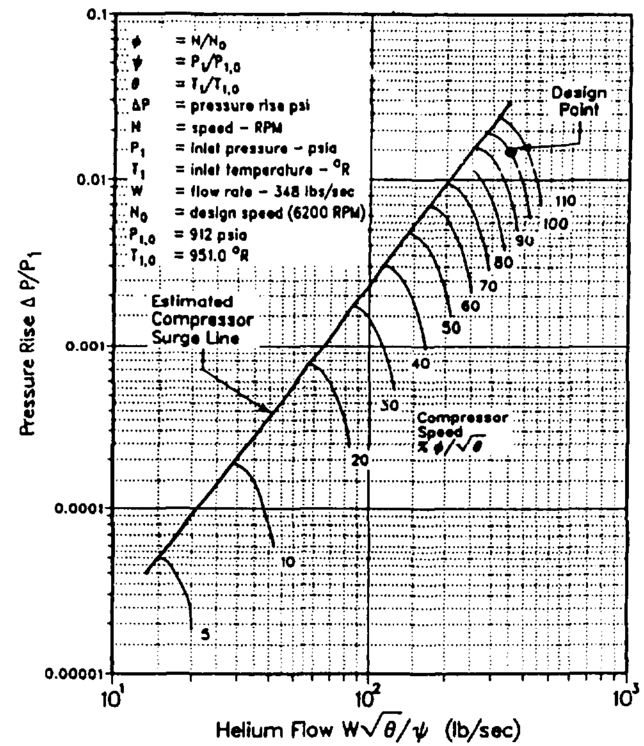


Fig 6. Main Circulator Compressor Map (Normal Pressurized Operating Condition)

factor exceeds 0.60 (Ref. 14). From Fig. 7 it can be seen that with a maximum rotor diffusion factor of 0.42 the design has conservative loading. Without exceeding the D value of the FSV compressor (which has good characteristics), the compressor could be rebladed within the same annulus and rotational speed to give about 17 psi rise (28% increase). The attendant power increase would, of course, necessitate a larger electric motor envelope. It is likely that by blade geometry/speed optimization a single-stage compressor could be designed for a pressure rise of over 20 psi, and indeed this was demonstrated for the Delmarva plant.

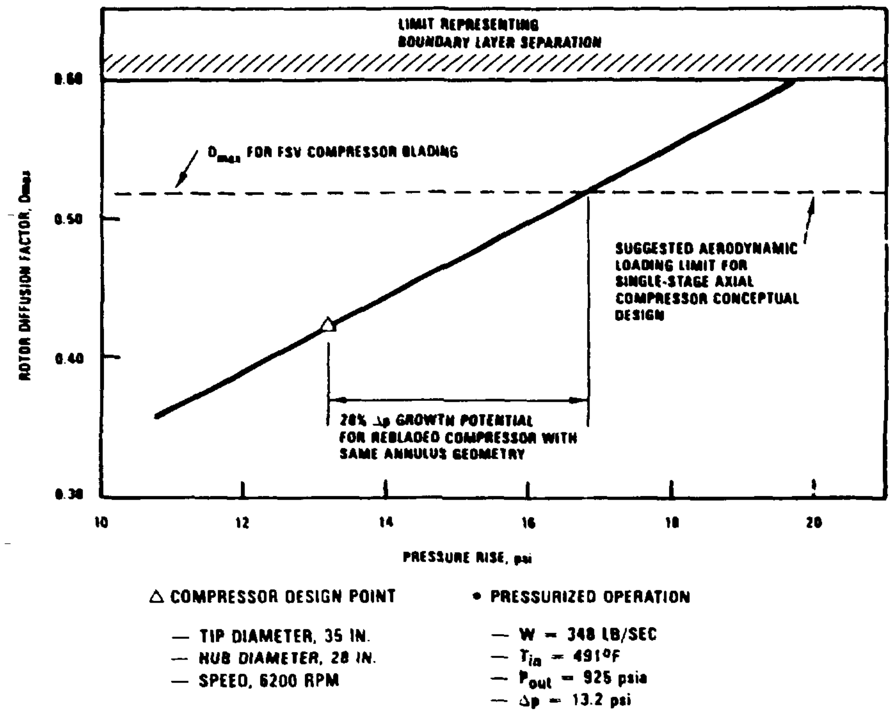
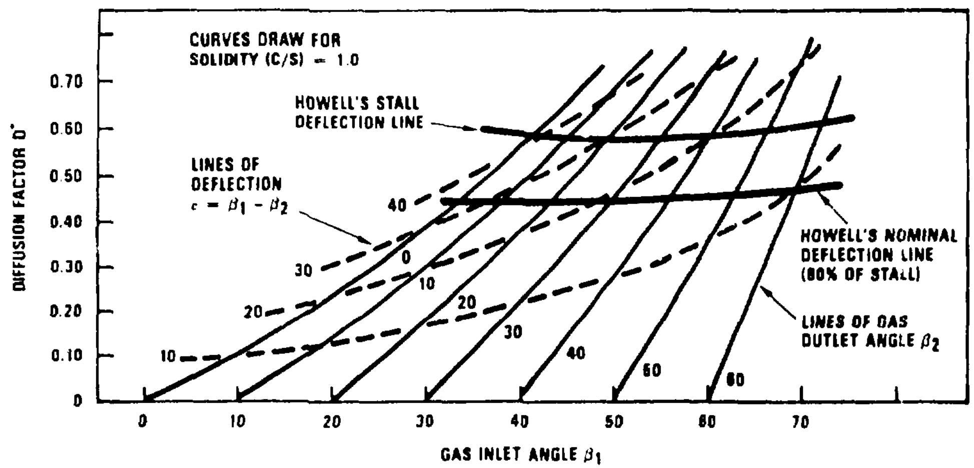


Fig. 7 Pressure Rise Growth Potential for Axial Compressor

In closing, it has been shown that a single-stage axial flow compressor has the capability to give a pressure rise of around 50% higher than the currently estimated circuit pressure loss of 13.2 psi.

3.4. REFERENCE COMPRESSOR DESIGN

Based on the aforementioned studies, the single-stage axial flow variant was selected, it being concluded that it best satisfied the requirements. A strong factor in this selection was the near commonality with the FSV compressor impeller, a view of which is shown on Fig. 9. Aerodynamic data for the MHTGR axial compressor blading is given on Table 5, and the major parameters and features are compared



*DEFINED AS $D = [1 - (\cos\beta_1/\cos\beta_2)] + S/C\,(\cos\beta_1/2)\,(\tan\beta_1 - \tan\beta_2)$

Fig. 8 Comparison of Howell's (U.K.) and NASA Diffusion Factor Methods of Defining Axial Compressor Aerodynamic Loading

with the FSV machine on Table 6. An overall view of the circulator (discussed in Ref. 1) is shown on Fig. 10. Again, using a scoping criteria discussed earlier, it can be seen from Fig. 11 that the proposed design is in the regime of high efficiency for an axial flow compressor. Major parameters for the selected machine are given on Table 7.

A major change from an earlier design variant (Ref. 15) involved a change in the gas flow direction through the impeller. In the reference design (Fig. 10), the helium flow is downwards through the compressor blading, and the upward aerodynamic thrust (4500 lb) partially offsets the downward rotor weight (6500 lb) to ease the requirements on the catcher thrust bearing. In the previous design, as would be the case for a radial flow impeller concept (as in Fig. 2), the thrust and weight would be additive, this resulting in more demanding requirements on the catcher bearing.

Fig. 9 Fort St. Vrain Circulator Rotating Assembly

TABLE 5. AERODYNAMIC DATA FOR AXIAL COMPRESSOR BLADING

RADIAL POSITION	ROOT	TIP
IMPELLER DIAMETER, mm (IN.)	711 (28.0)	889 (35.0)
HUB/TIP RATIO	0.80	
AXIAL VELOCITY, m/SEC (FT/SEC)	125 (411)	125 (411)
BLADE SPEED, m/SEC (FT/SEC)	231 (758)	289 (947)
FLOW COEFFICIENT, Va/U	0.54	0.43
ROTOR BLADE CHORD, mm (IN.)	82.0 (3.23)	65.8 (2.59)
STATOR BLADE CHORD, mm (IN.)	64.3 (2.53)	64.3 (2.53)
ROTOR SOLIDITY, C/S	1.28	0.82
BLADE HEIGHT, mm (IN.)	88.9 (3.5)	
ROTOR ASPECT RATIO	1.20	
ROTOR MACH NUMBER	0.19	0.23
ROTOR DIFFUSION FACTOR	0.40	0.33
STATOR DIFFUSION FACTOR	C.45	0.41
DEGREE OF REACTION	0.81	0.88
MEAN ROTOR INLET ANGLE	64.3	64.3
MEAN ROTOR OUTLET ANGLE	55.6	55.6
MEAN STATOR INLET ANGLE	31.6	31.6
MEAN STATOR OUTLET ANGLE	0	0
MEAN ROTOR STAGGER ANGLE	56.4	56.4
MEAN STATOR STAGGER ANGLE	12.4	12.4
NUMBER OF ROTOR BLADES	35	
NUMBER OF STATOR BLADES	37	
BENDING STRESS, MPa (psi)		
ROTOR BLADE	34.7 (5,000)	
STATOR BLADE	34.7 (5,000)	
ROTOR CENTRIFICAL STRESS, MPa (psi)	70.2 (10,110) MAX.	
COMPRESSOR IMPELLER EFFICIENCY, %	88.75	
DIFFUSER EFFICIENCY, %	80.0 (0.24 AREA RATIO)	
OVERALL EFFICIENCY (INCLUDING DIFF), %	79.2	

NOTES: 1) ROTOR BEFORE STATOR (ZERO INLET AND EXIT SWIRL).
2) NASA 65 SERIES PROFILE
3) DESIGN METHODOLOGY AS FOR FSV COMPRESSOR.

4. SHUTDOWN COOLING CIRCULATOR

4.1. INSTALLATION

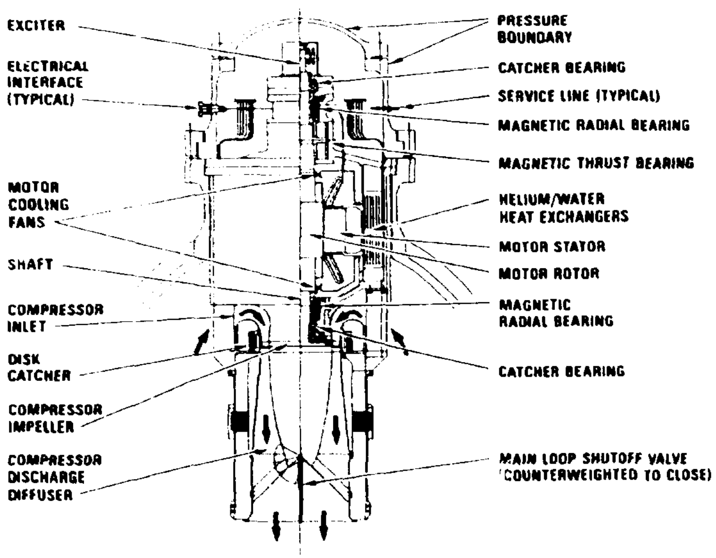
As shown on Fig. 12, the shutdown cooling circulator is installed in the bottom head of the reactor vessel. This small circulator (not safety-related) is used to provide rapid cooling of the reactor system (for refueling, maintenance, repair, etc.) if the main loop is unavailable. The system is capable of decay heat removal in pressurized and depressurized modes of operation.

4.2. REQUIREMENTS

Comprehensive requirements for the circulator have been discussed previously (Ref. 2), and boundary conditions necessary for the design of the circulator are given in Table 8.

TABLE 6. HELIUM CIRCULATOR FEATURES COMPARISON

FEATURE/PARAMETER	FORT ST. VRAIN	MHTGR
HELIUM FLOW, Kg/SEC (LB/SEC)	126 (278)	158 (348)
INLET TEMPERATURE, °C (°F)	394 (742)	255 (491)
INLET PRESSURE, MPa (PSIA)	4.73 (686)	6.29 (912)
PRESSURE RISE, KPa (PSID)	96.5 (14.0)	91 (13.2)
ADIABATIC HEAD, M (FT)	2885 (9466)	1918 (6309)
CIRCULATOR ORIENTATION	VERTICAL	VERTICAL
COMPRESSOR TYPE	AXIAL	AXIAL
COMPRESSOR DRIVE	STEAM TURBINE	ELECTRIC MOTOR
FLOW CONTROL	VARIABLE SPEED	VARIABLE SPEED
SPEED, RPM	9550	6200
POWER, KW(e) (HP)	3952 (5300)	3169 (4250)
TORQUE, M-Kg (FT-LB)	406 (2930)	488 (3600)
COMPRESSOR EFFICIENCY, %	80.0	79.2
IMPELLER TIP DIAMETER, MM (INS)	711 (28)	889 (35)
BLADE HEIGHT, MM (INS)	121 (4.76)	88.9 (3.5)
NUMBER OF ROTOR BLADES	31	35
TIP SPEED, M/SEC (FT/SEC)	368 (1187)	289 (947)
SHAFT DIAMETER, MM (IN)	165 (6.5)	222 (8.75)
ROTOR WEIGHT, Kg (LB)	273 (600)	2955 (6500)
BEARING TYPE	WATER-LUBRICATED	ACTIVE MAGNETIC BEARINGS
BEARING SPAN, MM (IN)	559 (22)	2337 (92)
BEARING CAPACITY		
RADIAL/BEARING, Kg (LB)	1591 (3500)	2045 (4500)
THRUST, Kg (LB)	5455 (12,000)	5455 (12,000)
FIRST CRITICAL SPEED, RPM (%)	13,000 (145)	7600 (123)
MISSILE PROTECTION CAPABILITY	YES	YES
SAFETY CLASS	SAFETY CLASS	NON-SAFETY
BACKUP DRIVE	PELTON WHEEL	NA
MACHINE INSTALLATION	IN PCRV (BELOW CORE)	TOP OF STEAM GENERATOR VESSEL
MACHINE STATUS	OPERATIONAL (>250,000 HOURS)	CONCEPTUAL DESIGN



for MHTGR

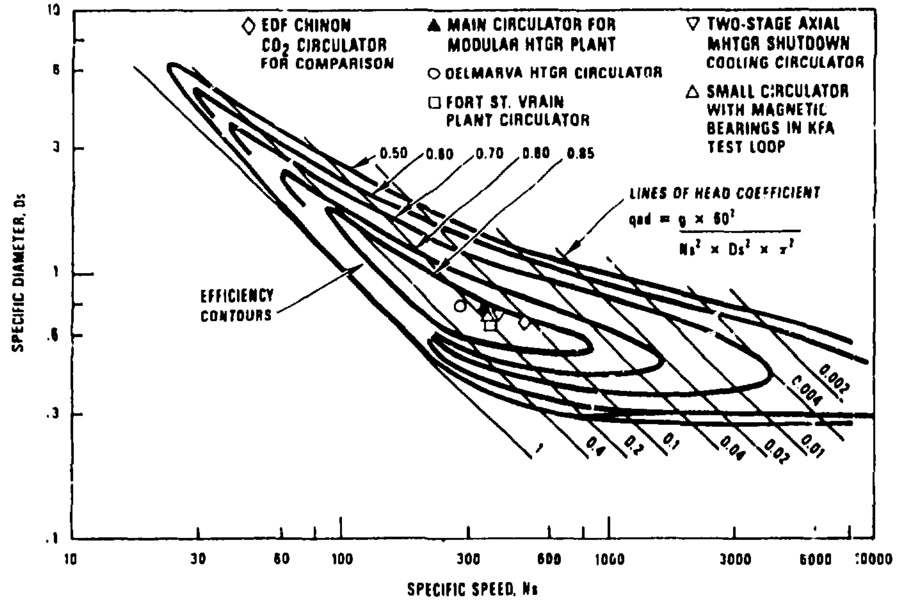


Fig. 11 Specific Speed - Diameter Array for Axial Flow Compressor

TABLE 7. MAJOR PARAMETERS FOR SELECTED HELIUM CIRCULATOR

- SINGLE STAGE AXIAL FLOW COMPRESSOR
- IMPELLER DIAMETER 35 IN. (889 mm) } BLADE GEOMETRIES VERY SIMILAR TO FSV COMPRESSOR
- ROTATIONAL SPEED 6200 RPM
- TIP SPEED 947 FT/SEC (289 m/SEC)
- CONSERVATIVE STRUCTURAL DESIGN
- POWER 4250 HP (3169 kW(e))
- CONSERVATIVE AERODYNAMIC LOADING COULD ACCOMMODATE UP TO 25% INCREASE IN CIRCUIT RESISTANCE
- GOOD SURGE MARGIN OVER WIDE FLOW RANGE
- CONCEPTUAL DESIGN CLOSE TO OPTIMUM FOR MAXIMUM EFFICIENCY
- OVERALL MACHINE DIAMETER 8.0 FT (2.4 m)
- OVERALL MACHINE LENGTH 20.0 FT (6.1 m)
- ASSEMBLY WEIGHT 30 TONS (27,220 Kg)
- MACHINE ASSEMBLY CAN BE READILY REMOVED AND REPLACED IN SPACE ABOVE STEAM GENERATOR VESSEL
- DESIGN BASED ON EXISTING AND PROVEN TECHNOLOGY
- EXTENSIVE INDUSTRY EXPERIENCE FOR COMPRESSOR DESIGN AND FABRICATION

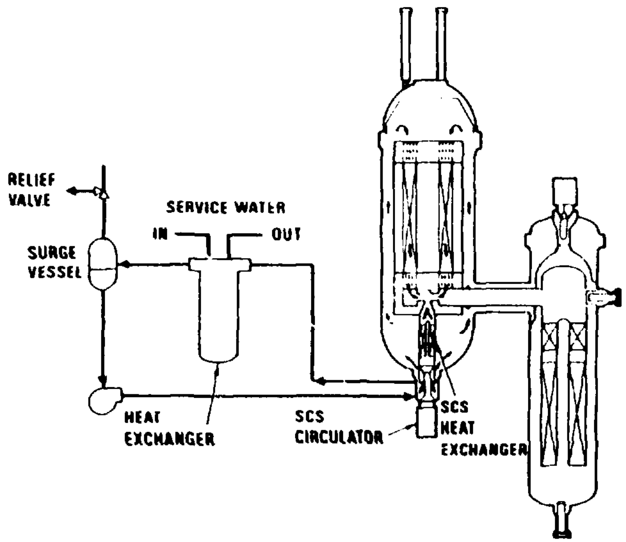


Fig. 12 Small Circulator Installed in Shutdown Cooling Loop

TABLE 8. SHUTDOWN COOLING CIRCULATOR DESIGN REQUIREMENTS

- PROVIDE HELIUM FLOW OF 6.36 LB/SEC (2.89 Kg/SEC)
- PROVIDE HELIUM PRESSURE RISE (DEPRESSURIZED) OF 0.71 PSI (4.9 kPa)
- PROVIDE CAPABILITY TO CIRCULATE HELIUM FOR DEPRESSURIZED AND *PRESSURIZED SYSTEM CONDITIONS*
- FLOW CONTROL BY VARIABLE SPEED MACHINE
- ENSURE STABLE OPERATION OVER WIDE SPEED RANGE (5%-100%)
- INCORPORATE PRESSURE RISE GROWTH CAPABILITY
- INSTALL CIRCULATOR IN LOW TEMPERATURE PART OF CIRCUIT
- BASE DESIGN ON ESTABLISHED AND PROVEN TECHNOLOGY
- MINIMIZE MACHINE ENVELOPE TO FACILITATE REMOVAL AND REPLACEMENT IN SPACE BELOW REACTOR VESSEL
- STRESS LEVELS TO BE COMMENSURATE WITH 40 YEAR LIFE REQUIREMENT

4.3. AXIAL VERSUS RADIAL FLOW COMPRESSOR AERODYNAMIC CONSIDERATIONS

A summary of the major factors in the selection of the impeller type are given on Table 9. Since the circulator is located near the bottom of the below-grade silo (Ref. 1), a dominant consideration in the early phase of the design was to minimize the machine size (particularly the diameter, since this impacts the size of the removal cask). Initial focus was on a high-speed axial flow machine, and a design was established. This design was not viewed as being conservative and would involve considerable extrapolation from the FSV data base; accordingly, studies of a lower-speed radial compressor were undertaken, and various aspects of the resulting comparison are given in the following sections.

4.3.1. Axial Flow Impeller

In reviewing the machine operating envelope, it was quickly determined that the depressurized mode (with very low gas density) gave the

TABLE 9. AXIAL OR RADIAL COMPRESSOR FOR SHUTDOWN COOLING CIRCULATOR

CONSIDERATION	AXIAL FLOW COMPRESSOR	RADIAL FLOW COMPRESSOR
DESIGN STATUS	INITIAL DESIGN CONCEPT	CONCEPTUAL DESIGN
MEETING DESIGN PRESSURE RISE REQUIREMENT (0.71 psi)	2 AXIAL STAGES NEEDED	SINGLE STAGE
PRESSURE-RISE MARGIN CAPABILITY (25% ABOVE DESIGN)	YES, WITH 2 STAGE	YES
EXPERIENCE/TECHNOLOGY	*EXTRAPOLATION FROM SINGLE-STAGE FSV MACHINE*	*INDUSTRIAL MACHINE*
PERFORMANCE	ALTHOUGH NOT OPTIMIZED POWER ABOUT 200 HP (150 kW(e))	CONCEPT WILL NEED SLIGHT INCREASE IN POWER
SIMPLICITY/RUGGEDNESS	HAVING 2 STAGES ADDS COMPLEXITY	SIMPLE ROTOR
DESIGN CONSERVATISM	HIGH SPEED MACHINE WITHIN STRESS LIMITS	LOWER SPEED, MORE CONSERVATIVE CONCEPT
MEETS SYSTEM REQUIREMENTS	YES	YES
MACHINE ENVELOPE	ASSEMBLY DIAMETER MINIMIZED FOR EASE OF REMOVAL	1 FOOT (0.3 m) INCREASE ON ASSEMBLY DIAMETER ACCEPTABLE
OVERALL SUMMARY	DESIGN MEETS REQUIREMENTS EXTRAPOLATION FROM EXISTING DATA BASE NOT CONSERVATIVE DESIGN	SIMPLER OVERALL MACHINE CONSERVATIVE PARAMETERS SLIGHTLY INCREASED ENVELOPE

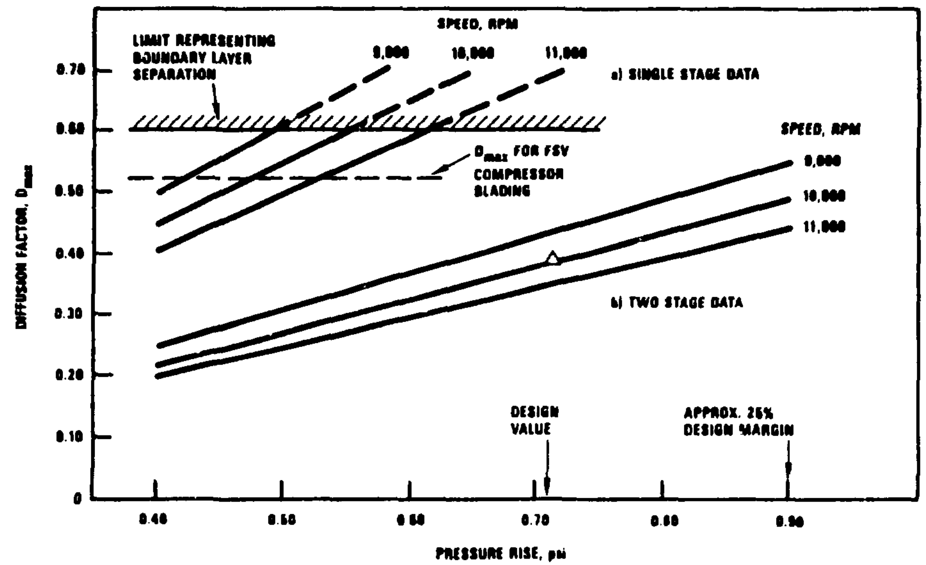
most severe compressor requirement in terms of adiabatic head as shown on Table 10. With a head rise on the order of 50% greater than for the FSV machine, it became clear that the design pressure rise requirement of 0.71 psi could not be realized with a single-stage axial flow compressor. The impact of number of stages on the blade loading factor (D) is shown on Fig. 13. A two-stage design was selected based on an impeller diameter of 737 mm (29 in.) and a rotational speed of 10,000 rpm. With a diffusion factor of around 0.4, the blade loading is very conservative. The resultant specific speed and diameter of 381 and 0.75, respectively, gives a machine in the island of maximum efficiency as shown on Fig. 11. A layout of the circulator assembly embodying a two-stage axial flow compressor is given on Fig. 14.

4.3.2. Radial Flow Impeller

As a starting point in the study, an observation of Fig. 3 would indicate that to be in the center of the maximum efficiency island, the

TABLE 10. COMPRESSOR COMPARISON FOR SHUTDOWN COOLING CIRCULATOR

COMPRESSOR TYPE	AXIAL FLOW			RADIAL FLOW	
COMPRESSOR DESIGN	MHTGR 2 STAGE DESIGN	MHTGR 1 STAGE CONCEPT	FSV MACHINE (FOR COMPARISON)	MHTGR CONCEPT	THTR CIRCULATOR (FOR COMPARISON)
MACHINE STATUS	CONCEPTUAL DESIGN	DATA POINT	OPERATIONAL	CONCEPTUAL DESIGN	OPERATIONAL
NUMBER OF STAGES	2	1	1	1	1
PRESSURE RISE, psi	0.71	0.50	14.0	0.71	18.0
HELIUM FLOW, LB/SEC	6.36	6.36	278	6.36	112
INLET TEMP, °F	240	240	742	230	482
OUTLET PRESSURE, psia	14.0	14.0	700	14.0	569
INLET DENSITY, LB/FT^3	0.00708	0.00708	0.213	0.00708	0.218
VOLUMETRIC FLOW, FT^3/SEC	898	898	1304	898	514
ADIABATIC HEAD, FT	14,500	10,170	9465	14,500	11,890
IMPELLER O/D, IN.	29.0	29.0	28.0	38.0	35.4
SPEED, RPM	10,000	10,000	9550	5400	5600
TIP SPEED, FT/SEC	1265	1265	1167	895	865
SPECIFIC SPEED, N_s	381	295	360	123	111
SPECIFIC DIAMETER, D_s	0.75	0.81	0.64	1.22	1.35
POWER, HP	207	150	5300	220	3083
PRESSURE RISE CAPABILITY	SINGLE STAGE AXIAL LIMITED TO ABOUT 0.5 psi PRESSURE RISE TWO STAGE AXIAL HAS CAPABILITY TO 0.90 psi (APPROX. 25% MARGIN ABOVE DESIGN VALUE)			LOW SPEED RADIAL HAS CAPABILITY TO GIVE HIGH PRESSURE RISE (>0.9 psi)	



△ DESIGN POINT DATA FOR TWO STAGE AXIAL FLOW SHUTDOWN COOLING CIRCULATOR
— SYSTEM DEPRESSURIZED
— Δp = 0.71 psi
— N = 10,000 RPM
— D_{tip} = 29 IN.

Fig. 13 Axial Flow Compressor Aerodynamic Loading

specific speed and diameter should be on the order of 100 and 1.5, respectively. An example of a machine in this regime is the radial flow compressor for the THTR plant circulator.

For the design valve of pressure rise (i.e., 0.71 psi), an effort was expended to investigate the impact of major parameters on a radial flow compressor, and data are portrayed on Fig. 15. Without regard to envelope constraint, the first iteration in the design process would be to select specific speed and diameter values consistent with operation in a high-efficiency regime. This would yield a low-speed machine [e.g., 4400 rpm with a 1194-mm (47-in.) diameter impeller], with a machine envelope much greater than the aforementioned axial flow variant.

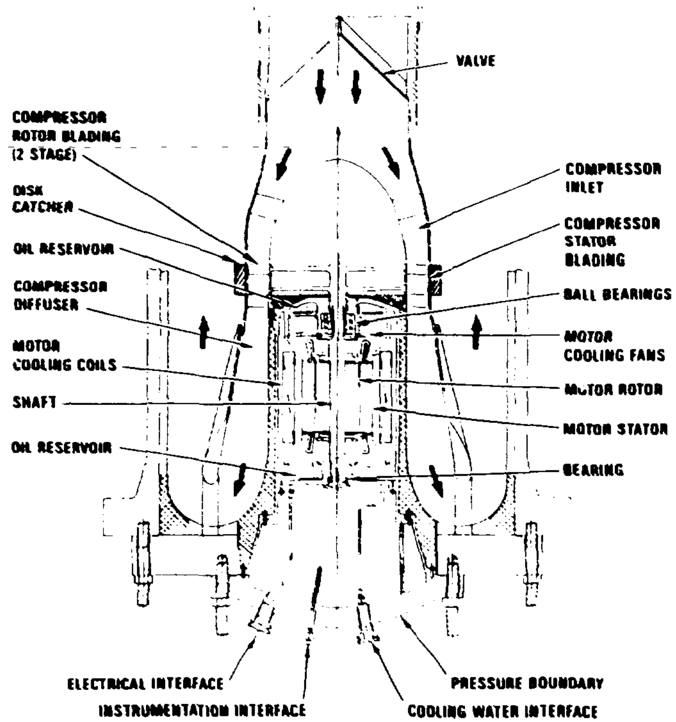


Fig. 14 Initial 2 Stage Axial Flow Shutdown Cooling Circulator Concept

For concept layout purposes a machine was selected with an impeller diameter of 965 mm (38 in.) and a rotational speed of 5400 rpm. From Fig. 15 it can be seen that this selection is still within the acceptable envelope, although slightly outside of the maximum efficiency island (Fig. 3). Such a radial compressor has the potential for at least a 25% in pressure rise.

4.4. REFERENCE COMPRESSOR DESIGN

There were essentially two factors that led to the selection of the radial flow machine: (1) the desire to avoid using a high tip speed,

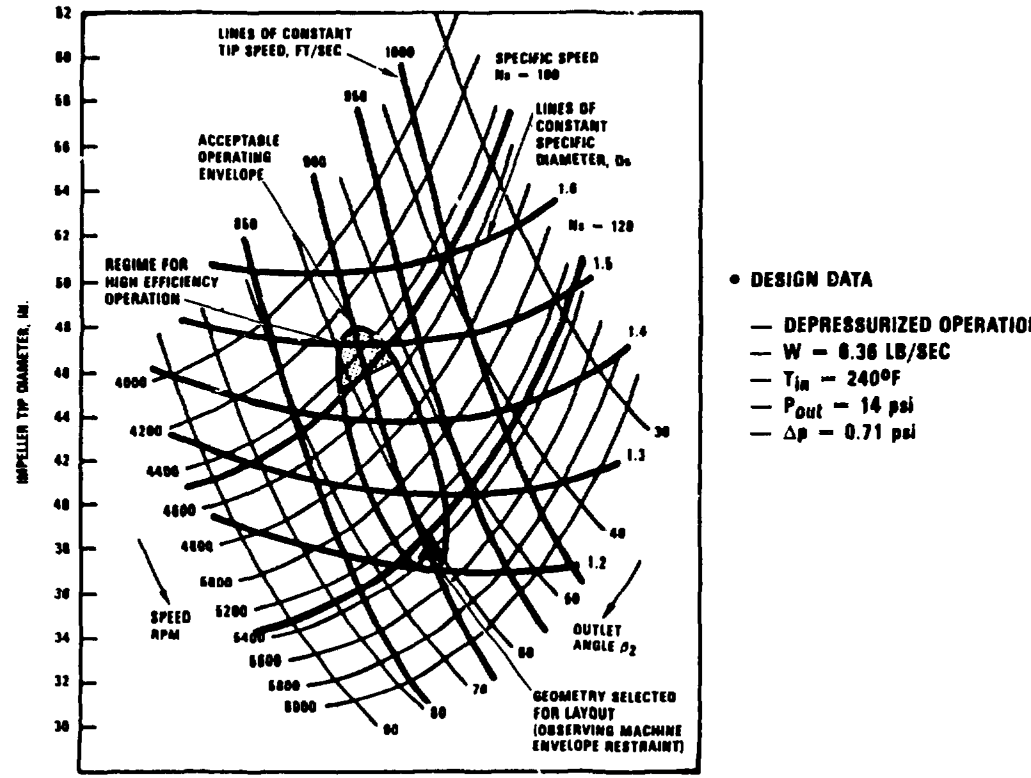


Fig. 15 Parametric Array for Radial Compressor

two-stage axial compressor (since it involves considerable extrapolation from the proven FSV impeller), and (2) it takes full advantage of the good European experience with radial flow compressors in gas-cooled reactors.

Detailed aerodynamic design of the radial flow compressor remains to be done, but sufficient analyses and design work have been performed to identify the major parameters and features. A representative layout of the circulator is shown on Fig. 16. Striving for maximum efficiency

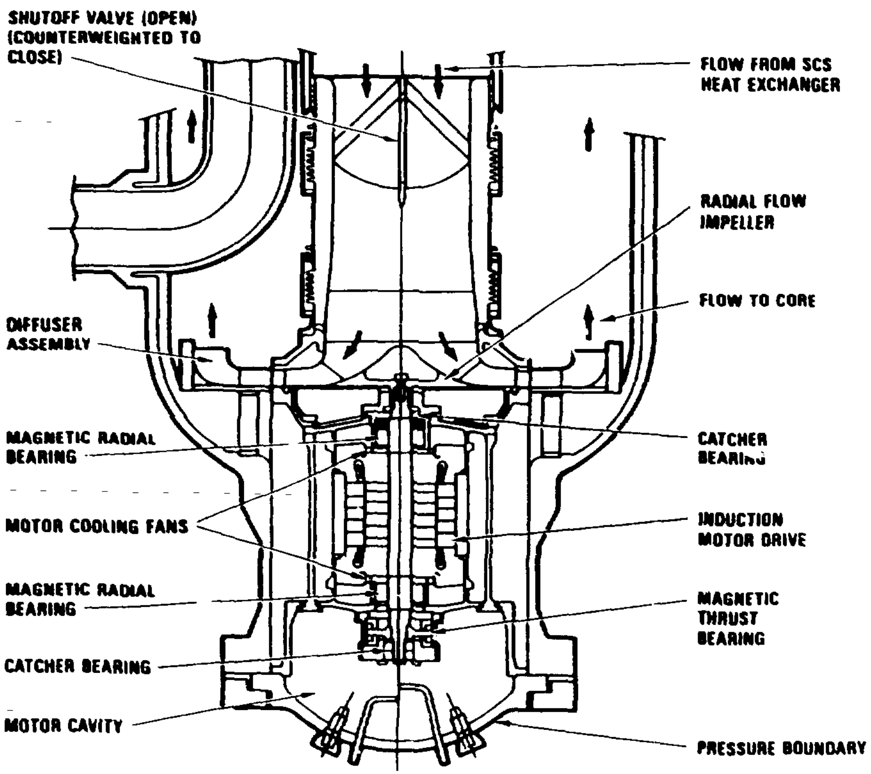


Fig. 16 Radial Flow Shutdown Cooling Ci :culator Concept

is not a strong requirement for this machine, since its service will be intermittent, and will be called upon to operate in the rare event that the main loop is unavailable. Major parameters for the selected machine are given on Table 11.

TABLE 11. MAJOR PARAMETERS FOR SHUTDOWN COOLING CIRCULATOR

- RADIAL FLOW COMPRESSOR
- IMPELLER DIAMETER 38.0 IN. (965 mm) } VERY SIMILAR TO THTR PLANT CIRCULATOR
- ROTATIONAL SPEED 5400 RPM
- TIP SPEED 895 FT/SEC (273 m/SEC)
- CONSERVATIVE STRUCTURAL DESIGN
- PCWER 220 HP (164 kW(e))
- CONSERVATIVE AERODYNAMIC LOADING COULD ACCOMMODATE UP TO 25% INCREASE IN CIRCUIT RESISTANCE
- DESIGN NOT OPTIMIZED FOR MAXIMUM EFFICIENCY (MACHINE USE WILL BE INTERMITTENT DURING PLANT LIFE) BUT RATHER FOR MINIMUM ENVELOPE
- OVERALL MACHINE DIAMETER 4 FT (1.23 m)
- OVERALL MACHINE LENGTH 9.6 FT (2.93 m)
- MACHINE ASSEMBLY CAN BE READILY REMOVED AND REPLACED IN SPACE AVAILABLE BELOW REACTOR VESSEL
- DESIGN BASED ON EXISTING AND PROVEN TECHNOLOGY
- EXTENSIVE INDUSTRY EXPERIENCE FOR COMPRESSOR DESIGN AND FABRICATION

5. SUMMARY

In surveying the over 200 circulators that are operational (or have operated) in gas-cooled reactors, a variety of impeller types (radial, mixed, and axial flow) and drives (electric motor, steam turbine) can be observed. In the case of applications involving the pumping of dense carbon dioxide (i.e., AGR plants), or helium in HTRs with a high circuit resistance (i.e., pebble bed reactor), the choice of a radial flow compressor is quite clear. This paper has addressed the choice of compressor type for the MHTGR with a low circuit resistance characteristic of the prismatic core. The following results were obtained from the circulator design studies:

- Axial Flow Compressor Well Suited for Main Circulator
  - Impeller similar to FSV compressor which performs well.
  - Near or optimum for maximum efficiency.

- Has pressure rise growth potential.
- Well suited to pressure loss in prismatic reactor.
- Conservative aerodynamic and structural loading.
- Simple machine with no variable geometry features.
- Good surge margin over wide flow range.
- Selected flow direction through impeller results in thrust loading partially offsetting rotor weight (eases thrust bearing requirements).

• Radial Flow Compressor Ideal for Shutdown Cooling Circulator

  - Established machine envelope facilitates ease of removal/ replacement.
  - Not optimized for maximum efficiency (machine has intermittent service).
  - Has pressure rise growth potential.
  - Simple straightforward small machine.
  - Conventional by industry standards.

The compressor aerodynamic designs were performed using established and validated computer codes. A high level of confidence exists that the compressors will perform as predicted. With the well-established technology base, the need for a compressor development program is not foreseen. A well-established vendor infrastructure exists for the design and fabrication of both machine types. The nomenclature is included on Table 12.

TABLE 12. COMPRESSOR AERODYNAMIC PARAMETERS/NOMENCLATURE

	C	BLADE CHORD		V	INLET VOLUMETRIC FLOW, FT^3/SEC
•	D	IMPELLER DIAMETER, FT	•	Va	AXIAL GAS VELOCITY, FT/SEC
•	D_s	SPECIFIC DIAMETER, D. $HAD^{0.25}/\sqrt{v}$		Va/U_m	FLOW COEFFICIENT
•	D_{max}	MAXIMUM DIFFUSION FACTOR		ρ	INLET GAS DENSITY, LB/FT^3
	Had	ADIABATIC HEAD, $\Delta P/\rho$ FT-LB/LB	•	Δp	PRESSURE RISE, $LB/IN.^2$, LB/FT^2
•	N	ROTATIONAL SPEED, RPM		qad	HEAD COEFFICIENT
•	N_s	SPECIFIC SPEED, $N \cdot \sqrt{v}/Had^{0.75}$		P_i	INLET PRESSURE, $LB/IN.^2$
	S	BLADE SPACING		W	MASS FLOW, LB/SEC
•	C/S	SOLIDITY		T_1	INLET TEMPERATURE, °F
•	U	BLADE TIP SPEED, FT/SEC		ΔH	ENTHALPY RISE, Btu/LB
	U_m	MEAN BLADE SPEED, FT/SEC		$\Delta H/U_m$	TEMPERATURE RISE COEFFICIENT
	g	ACCELERATION DUE TO GRAVITY, FT/SEC^2		J	MECHANICAL EQUIVALENT OF HEAT, FT-LB/Btu
				Cp	SPECIFIC HEAT, Btu/LB°F

*VARIABLE PARAMETERS IN AERODYNAMIC STUDIES

ACKNOWLEDGMENT

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MAIN GAS CIRCULATOR FOR VG-400 REACTOR PLANT

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Abstract

Principle parameters and operating conditions of the main gas circulator (MGC) in ВГ-400 reactor plant are presented.

Brief MGC design description and experimental work scope are given.

1. INTRODUCTION

Important design features of ВГ-400 reactor plant are integral lay-out of the main primary circuit components in a prestressed concrete reactor vessel and helium coolant use /1/. Main gas circulator is one of the key plant components, effecting the reactor lay-out and its long availability.

Four MGCs are allowed in the reactor plant. Circulators are high energy consumption components - up to 5% of the plant rating, due to high circulator capacity.

That is why the choice of the optimum circulator gas path configuration substantially effects both