EUROPEAN COMMISSION
5th EURATOM FRAMEWORK PROGRAMME 1998-2002
KEY ACTION : NUCLEAR FISSION

HTR-E

High-Temperature Reactor Components and Systems

CONTRACT N°
FIKI-CT-2001-00177

WP3 Magnetic bearings
Helium rotating seal State of the art and specifications

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Dissemination level : RE
Document N°: HTR-E-02/06-D-4-1-0
Status : Final
Deliverable n°30

FRAMATOME ANP

NOVATOME

NUMERO D'OTP OU EOTP	NOMBRE DE PAGES : 20	RAISON SOCIALE FOURNISSEUR :
	NOMBRE D'ANNEXES : 2	
12T.DTHTREW4NV	CONFIDENTIALITE : N	NUMERO DE COMMANDE / IMMATRICULATION FOURNISSEUR

CODE		DESIGNATION
AFFAIRE	H T	HTR
NATURE	D C	Note technique
THEME	C C P	Conception
MATERIEL	4 2 0 0	Bloc réacteur - Généralités

TITRE : **Helium rotating seal. State of the art and specifications**

TITRE ABREGE : **H E L I U M R O T A T I N G S E A L**

IND	DATE	ETAT	REDACTEUR NOM-VISA	VERIFICATEUR NOM-VISA	APPROBATEUR NOM-VISA	MODIFICATIONS – OBSERVATIONS – ACCORDS REFERENCES ET DATES
A	04/06/2002	PRE	B.ROSSIN	E.BREUIL	J.ROLEAU	First issue

IMMATRICULATION CLIENT :

IMMATRICULATION INTERNE :	EVEK	DC	02	0250	/
	UNITE	TYPE	ANNEE	CHRONO	TRIGRAMMES

10, rue Juliette Récamier 69456 LYON CEDEX 06 - Téléphone : 04.72.74.82.21 - Télécopie : 04.72.74.84.90

Modifications

Revision	Date	Modifications
A	04/06/2002	First issue

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(1) INTERNATIONAL GT-MHR PROJECT
CONCEPTUAL DESIGN REPORT

1. Introduction

The GT-MHR and PBMR turbomachinery configurations are shown in appendix 1 and 2. The GT-MHR design features a submerged generator operating in a helium environment. While this concept results in a compact package, there are issues concerning the use of high pressure helium as a generator coolant and restricted accessibility to the generator.

Among the concerns are:

- electrical penetrations into the high pressure helium vessel
- effectiveness of helium as a generator coolant
- accessibility for generator maintenance

Removal of the generator from the pressure vessel will allow the use of a standard generator and eliminate problems associated with the high pressure helium and accessibility. The major obstacle to implementing removal is the viability of a suitable seal between the power conversion vessel and the rotating shaft that would connect the turbocompressor to the external generator.

For this configuration to be possible (see appendix 2 for PBMR configuration with rotating seal), the seal leakage must be held to stringent limits since the leakage consists of primary coolant helium with radioactive contamination.

2. Specifications and requirements for GT-MHR type rotating shaft seal

The operating environment for the shaft is sum up on the table below:

Requirement	Design point
Operating Fluid	Dry helium
Operating pressure	26 bars
Operating Temperature (towards leakage rate)	110°C
Design Temperature (towards mechanical design)	150°C
Emergency mode Temperature (short time)	450 to 500°C
Operating speed	3000 rpm
Trip speed possibility	3600 rpm
Shaft diameter	450 mm at least
Axial gap	0.5 + about 10 mm =10.5 mm
Radial gap	0.4 mm
Seal life	60 years (with inspections and parts replacements)
Leakage rate (of the entire reactor vessel)	55 g/hr equivalent to 0.31 Nm3/hr

The seal must restrict leakage of the hot helium in the vicinity of the turbine discharge at a pressure and temperature of 26 bars and about 110°C. The temperature of 150°C corresponds to the design temperature of the power conversion system vessel. The mechanical design temperature of the seal must indeed be the same temperature as the vessel. However, it must be kept in mind that, during emergency mode, the ambient medium temperature can reach 450 to 500°C for a short time.

In other respects, the seal must be suitable in an environment of dry helium with its gaseous impurities (H_2O < 2 vpm, CO_2 +CO < 6 vpm, H_2 < 5 vpm, and traces of Iodine and Xenon), solid graphite particle and gamma radiations.

The seal must successfully operate at 3000 rpm (±2%) on a 450 mm diameter shaft. The rotor is designed with a 20% safety margin which gives a 20 % overspeed possibility. It must withstand mechanical, acoustic and fluid dynamic vibrations resulting from turbomachine operation and fluid flow. It must be kept in mind that the seal must be operating in static conditions too (0 rpm..)

The shaft runout may be as high has 0.5 mm in the event of loss of magnetic levitation, rotor support is then provided by catcher bearings. A 0.4 mm radial displacement of the rotor must be taken into account for the same reason.

A relative axial displacement which could reach about 10 mm may be imposed on the seal due to differential thermal growths in the shaft between the rotating seal and the axial magnetic bearing.

Desirable life for the shaft seal is 60 years with inspections and parts replacements at regular intervals (as long as possible). The seal must also minimize technology needs. The seal assembly must be readily installed and removed and subcomponents of the seal easily replaced. Capability for testing prior to and during installation should be used as a criteria for feasibility.

In the case of shaft vibrations under various solicitations (earthquake, blade breakdown..), the speed of seal damages must be studied carefully in order not to cause a roughly decrease of the seal performances and a fast depressurization of the reactor vessel.

As a first assumption, a leakage of 10% of the reactor helium mass could be considered each year. It represents about 483 kg of helium each year which corresponds to a leakage rate of 55 g/hr, equivalent to 0.31 Nm3/hr (helium at 1 atm, 0 °C).

3. Specifications and requirements for PBMR type rotating shaft seal

The operating environment for the shaft is sum up on the table below:

Requirement	Design point
Operating Fluid	Dry helium
Operating pressure	26 bars
Operating Temperature (towards leakage rate)	110°C
Design Temperature (towards mechanical design)	150°C
Emergency mode Temperature (short time)	450 to 500°C
Operating speed	3000 rpm
Trip speed possibility	4200 rpm
Shaft diameter	350 mm
Axial gap	0.5 + about 10 mm =10.5 mm
Radial gap	0.4 mm
Seal life	40 years (with inspections and parts replacements)
Leakage rate (of the entire reactor vessel)	154 g/hr equivalent to 0.86 Nm3/hr

Except a few points, design requirements for PBMR rotating shaft seal are nearly the same as for GT-MHR configuration. The seal must be suitable in an environment of dry helium with its gaseous impurities : traces of H_2, CO, CH_4, N_2, H_3, I, Xe, Ar, Kr.

The seal must successfully operate at 3000 rpm (±2%), but on a 350 mm diameter shaft. The overspeed limitation of the shaft is 40 %, PBMR shaft is prone to speed trip due to the fact that the turbine shaft doesn't carry a compressor too. The reason for this particular choice is the extra safety margin that is available if the rotor is based on a standard 60 Hz rotor but used in a 50 Hz application. This standard 60 Hz rotor is designed with a 20% safety margin giving the 40 % overspeed possibility. It must withstand mechanical, acoustic and fluid dynamic vibrations resulting from turbomachine operation and fluid flow. It must be kept in mind that the seal must be operating in static conditions too (0 rpm..)

Desirable life for the shaft seal is 40 years with inspections and parts replacements at regular intervals (as long as possible). The seal assembly must be readily installed and removed and subcomponents of the seal easily replaced. Capability for testing prior to and during installation should be used as a criteria for feasibility.

As a first assumption, a leakage of 36.5 % of the reactor helium mass could be considered each year. It represents about 1350 kg of helium each year which corresponds to a leakage rate of 154 g/hr, equivalent to 0.86 Nm3/hr (helium at 1 atm, 0 °C). It is estimated that this leakage rate is low enough to allow the associated radioactive isotopes to disperse into the confinement, which would then be filtered and bled off to the atmosphere. Calculations on final design will confirm that dose rates to the public are negligible.

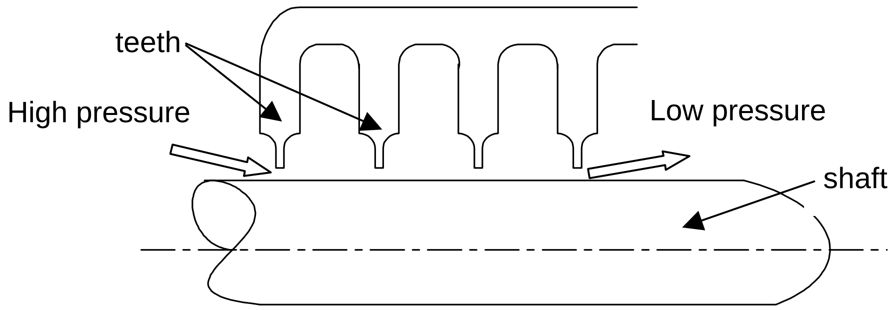
4. State of the art of dry gas rotating seal

4.1 Single shaft seals

Three categories of sealing configurations are selected for this state of the art description. Labyrinth seals, ring seals and face seals are commonly used in both aircraft mainshaft and industrial powerplant applications. Each of these sealing configurations are evaluated as a single shaft seal against the criteria listed in design requirements listed before.

4.1.1 Labyrinth seal

The primary mechanism for flow control in a labyrinth seal is the total loss of velocity head in the cavity formed between the teeth. The controlling flow equations are primarily a function of differential pressure, the number of teeth and the clearance between the inside diameter of the teeth and the journal diameter.



Labyrinth seal configuration

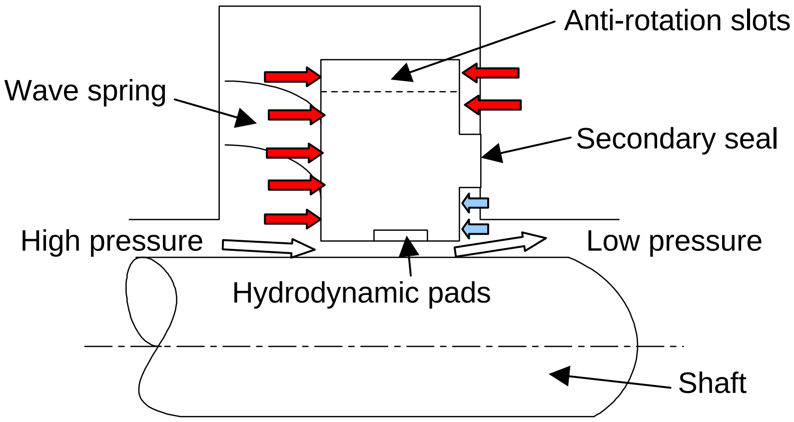
Assuming a single labyrinth seal with 40 teeth (more teeth didn't significantly improve the seal performance), an experimental leakage of helium on a 650 mm diameter shaft from the pressure vessel at 26 bars and about 500°C to an ambient environment is approximately 12000 Nm3/hr. (this value would be worse in our conditions, at 110°C, because the lower temperature: magnified density and reduced viscosity, results in a higher leakage rate). With an approximate tooth pitch of 5 mm, the extent of 40 teeth would require 200 mm of shaft length.

The labyrinth seal cannot be used as a single shaft seal due to its high leakage rate. This is primarily due to the excessively high (0.4 mm) clearance required by the radial catcher bearings. In all other respects, the labyrinth seal provides attractive characteristics:

- pressure and temperature are limited only by material capabilities
- high operational reliability
- unlimited life
- no size limitations
- no speed limitations

4.1.2 Dry gas ring seals

A dry gas ring seal is schematically described in figure below.



Dry gas ring seal configuration

The primary sealing element is the carbon ring which typically has a 0.025 mm diametrical clearance between its inside diameter and the journal diameter. A secondary sealing surface is shown on the side of the ring that prevents leakage around the outside diameter. A wave spring preloads the ring against the secondary sealing surface. Anti-rotation slots prevent the carbon ring from being rotated by viscous shear of the fluid between the ring and the rotating journal.

The arrows in the schematic denote regions of the ring that are under the influence of high pressures. By assessing the areas affected by these pressures, a force balance between the pressure loads and the wave spring load determine the amount of preload on the secondary sealing surface. Some adjustment of the pressure balance is available by changing the radial location of the secondary contact seal on the carbon ring.

Hydrodynamic pads are usually incorporated in the ring to aid in generating hydrodynamic forces that center the ring about the geometric center of the journal. Under ideal conditions, with the proper amount of pressure balance and spring load on the secondary surface, the ring can track the runout of the journal and maintain a continuous fluid film in the 0.025 mm clearance. If tracking does not occur, then contact will be made between the inside diameter of the ring and the journal. Wear from excessive contact will enlarge the clearance and result in excessive leakage.

Sliding contact is incurred at the secondary sealing surface as the carbon follows the motion of the journal. Preload on this sliding surface is critical to the acceptable operation of the seal. Too large a preload will not allow the ring to track the runout of the journal, too light a preload will allow excessive leakage through the secondary seal. In any case, some wear of the secondary sealing surface is unavoidable, limiting life of the carbon ring seal.

Carbon has been selected as the material of choice for the dry gas ring seal due to it's proven performance in contact seal applications.

From the foregoing general discussion of the geometry and function of a dry gas ring seal, the following observations can be made:

- the carbon ring seal geometry allows unlimited axial displacement capability

- occasional journal excursions to 0.4 mm allowed by the catcher bearings can be tolerated

- limited pressure balance capability of the carbon ring limits the allowable differential pressure

- wear is incurred at the contacting secondary sealing

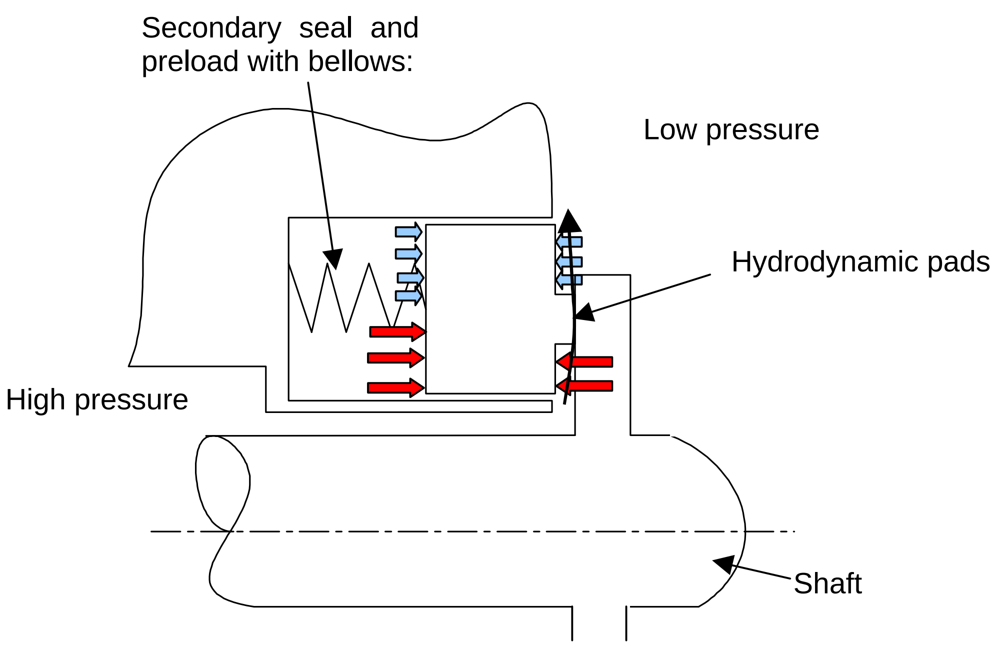
- surface contact with the journal may occur during off-design operations

- wear debris can be generated by the contacting secondary seal and journal contact during operations

- single seal leakage rate at design pressure of 26 bars and about 500°C is estimated to be about 80 Nm3/hr. (this value would be worse in our conditions, at 110°C, because the lower temperature: magnified density and reduced viscosity, results in a higher leakage rate)

4.1.3 Dry gas face seals

The dry gas face seal is similar in operation to the dry gas ring seal. The primary sealing surface is the axial face of a carbon ring against a rotor or collar attached to the shaft. This seal configuration is shown schematically in the figure below:



Dry gas face seal configuration

As in the dry gas ring seal, the sealing surface has hydrodynamic lift pads on either the carbon stator or rotating collar. The carbon stator is anti-rotated and spring loaded against the collar. The stator can form an effective static seal against the rotor provided that the two surfaces are flat enough. Surface flatness measure in terms of helium light bands are required to insure minimum leakage rates.

A secondary seal is required to prevent helium leakage around the back side of the carbon stator. The carbon stator is usually mounted in a metallic carrier to accommodate a variety of types of secondary seals. These include:

- 0-ring secondary seal
- piston ring seal
- a variety of spring energized static seals capable of some degree of movement
- bellows seal

The face seal in the figure above is shown with a bellows secondary seal. Due to the extended life requirements for this seal, the bellows seal was chosen as example as the representative secondary seal for the dry gas face seal. With a bellows secondary seal, the carbon stator face load and the secondary sealing function are provided by the same component. Since sliding contact at the secondary seal surface does not occur, leakage due to wear of the secondary seal contact surface is eliminated. The principles of operation for the dry gas face seal with other types of secondary seals are the same except for characteristics directly attributable to the bellows secondary seal. The estimate of a five year operating life for dry gas seals with other types of secondary seals can be used as an estimate for the operating life of the bellows seal. Limited life due to wear incurred on the primary sealing surface seems to be common for all dry gas face seals.

The disadvantages of bellows secondary seals are:

- possible life limit imposed by bellows stresses
- limited (but allowable) axial travel
- bellows dynamics must be considered in the system seal design

As rotation of the shaft begins, the initial motion of the rotor is accommodated by a sliding contact against the carbon stator. As speed increases, hydrodynamic pressure is built up that separates the sealing surface by a thin film of helium. For a 650 mm diameter shaft, it is estimated that an acceptable operating film thickness is approximately 0.013 mm. The balance between the static differential pressures acting on the carbon surfaces, the hydrodynamic pressure and the spring load determines the magnitude of the film thickness. Since the hydrodynamic pressure is a function of rotor speed, the film thickness can only be adjusted to a specific value of speed. Leakage may be acceptable at design speed, but the seal may not have sufficient performance at other speeds.

Low rotor speeds may allow the seal face to contact the rotor. This may cause abnormal wear or dynamic instability brought on by frictional effects. Higher rotor speeds may result in large leakage due to either large film thickness or dynamic instability generated by an unstable stator/rotor system.

A summary of dry gas face seals:

- Rotating collar must be attached to the shaft.

- High differential pressure capability is available due to the flexibility allowed for pressure balancing. Face seals up to 100 bars differential pressure have been demonstrated.

- Lifting surface on collar or stator generates a non-contacting gas film sealing surface.

- Limited axial displacement capability.

- Non-contact operation with low leakage is difficult to achieve

- 0.013 mm film thickness seems to be a practical value for a 650 mm diameter shaft.

- Single seal leakage rate at design pressure 26 bars and about 500°C on a 650 mm diameter shaft is estimated to be about 80 Nm3/hr. (this value would be worse in our conditions, at 110°C, because the lower temperature: magnified density and reduced viscosity, results in a higher leakage rate)

The higher performance capability of dry gas face seals comes at a cost of increased complexity and design considerations:

- The axial displacement must be carefully coordinated with the change in spring preload with displacement.

- Since the carbon stator preload is supported by the gas film, it is subjected to all the vagaries of a hydro-elastic dynamic systern.

- Hydrodynamic pressures are a strong function of the surface flatness since film thickness is very small. Thermal distortions must be considered to insure that the minimum flatness requirements are met for proper dynamic seal operation.

- Material compatibility is a significant factor to be considered. In view of the flatness requirement, any contact between the stator and the rotor must not generate unacceptable surface conditions detrimental to the development of the hydrodynamic film.

4.2 Single shaft seal summary

Since the film thickness and the diameter for both the dry gas ring seal and dry gas face seal are approximately the same, the estimated flow rates will be assumed to be the same (about 80 Nm3/hr with a 650 mm shaft diameter at 26 bars and about 500°C). Although differences in geometry are obvious, the overall influence on the computed flow rates are not significantly affected. The flow rate is primarily governed by viscous loss in the small clearance. The helium leakage is a function of film thickness. Therefore, it is apparent that helium leakage requirements cannot be satisfied with a single dry gas face ring or face seal.

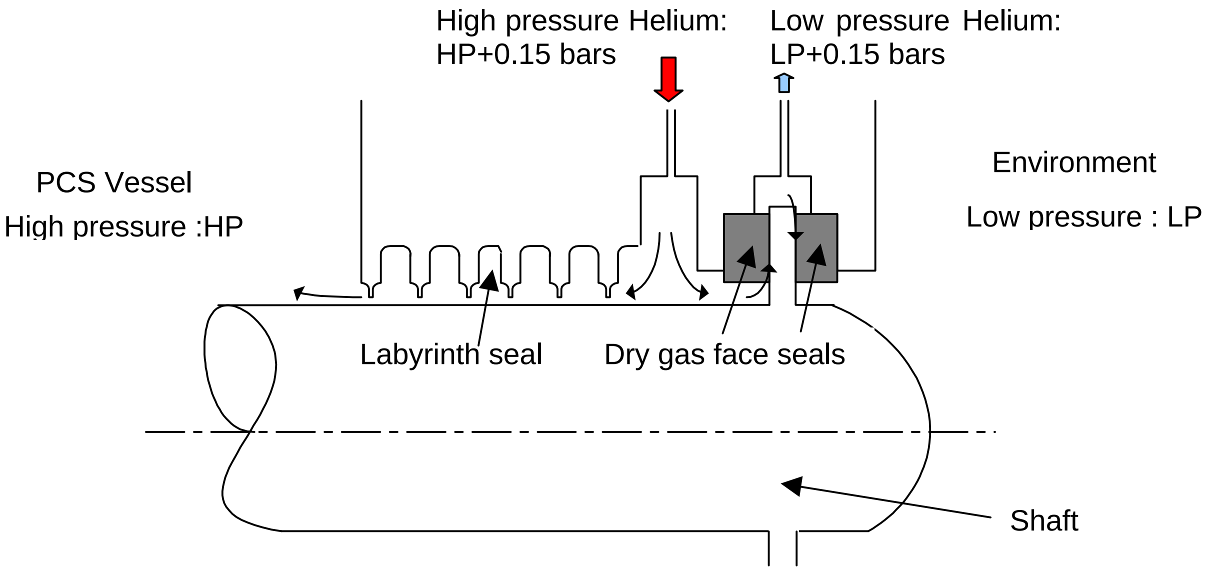
The labyrinth seal seemingly satisfies all of the criteria other than leakage but has an extremely high leak rate (about 12000 Nm3/hr with a 650 mm shaft diameter at 26 bars and about 500°C). The dry gas ring seal suffers from limited differential pressure capability and the dry gas face seals from limited axial displacement capability. Both the dry gas ring and face seal will require a test program to evaluate currently available material couples and potential candidate pairs. Neither the dry gas ring seal nor the dry gas face seal have been demonstrated in a helium atmosphere at a temperature, pressure and size required for this application.

Leakage flow analysis was carried out for multiple seals in series. Only the dry gas ring and face seals were considered for this analysis since it was clear that even multiple labyrinth seals would not satisfy the seal leakage limitations. Four dry gas ring or face seals in series resulted in a helium leak rate of 38 Nm3/hr (with a 650 mm shaft diameter at 26 bars and about 500°C). Therefore, single or multiples of the labyrinth, ring or face seals will not satisfy the allowable leakage rate of contaminated helium into the atmosphere. This conclusion would be the same in our conditions, at 110°C, because the lower temperature: magnified density and reduced viscosity, results in a higher leakage rate.

4.3 Buffered dry gas seal

4.3.1 Seal concept

The buffered seal concept utilizes a combination of seals to insure that the PC vessel helium leakage is minimized by controlling the leakage of an alternate source of helium. This sealing configuration involves a considerably more complex seal arrangement using the same basic seals discussed previously. The figure below shows a schematic of a possible combination of seals that may be utilized to provide a buffered seal system.



Buffered dry gas seal configuration

The PC vessel helium is prevented from leaking out of the PC vessel by a flow of high pressure buffer helium into the PC vessel. A 0.15 bars pressure drop was assumed to be sufficient to manifold the helium uniformly around the circumference of the labyrinth seal and prevent reverse flow of PC vessel helium into the buffering helium cavity.

The labyrinth seal is chosen as the primary seal for the buffered seal system since it has the highest reliability of any of the seals under consideration. Its only drawback is its excessively high leak rate, even though this rate will be considerably lower than for the single labyrinth seal described previously. With a 0.15 bars differential pressure, the leak rate into the PC vessel of the buffered helium is estimated to be 1500 Nm3/hr (with a 650 mm shaft diameter at 26 bars and at 500°C).

The high pressure helium must be contained by a secondary seal which has the same excessive leakage previously considered in the case of a single shaft seal. This seal leakage is contained by a low pressure helium scavenge which recycles the helium. This low pressure scavenge is kept slightly above ambient pressure such that a small amount of helium is allowed to leak into the atmosphere. This small leakage minimizes air contamination of the helium that is recycled back into the system. Maintaining the low pressure helium at 0.15 bars above ambient results in meeting the maximum leakage rate requirement of clean helium to the atmosphere of less than 2.5 Nm3/hr.

The clean helium flow rate that is recycled has the same pressure drop as that considered earlier for the dry gas ring or face seal. The lower temperature and reduced viscosity results in a higher leakage rate than that computed earlier.

4.3.2 Buffered seal summary

The following summary pertains to the configuration consisting of a labyrinth primary seal and two dry gas secondary seals :

- The labyrinth seal was selected for the primary seal due to its high temperature capability and inherent reliability. Since there are no contacting surfaces, there is no wear induced, debris nor wear induced degradation of performance. If the leak rates of clean helium into the vessel are too high then either a dry gas ring or dry gas face seal may be used in place of the labyrinth for the primary seal. However, the life of dry gas seals is likely to be limited, there are increased operational risks due to the limitations of these components, and there are technology risks associated with expanding the operational range of these seals.

- Helium was selected as a high pressure buffer fluid due to its acceptability in injecting it into the PC vessel helium. Since a certain percentage of the PC vessel helium was to be cleansed at given intervals, the clean helium could easily be injected as the buffer high pressure helium.

- A face seal was selected as the high pressure secondary buffer seal due to its high differential pressure capability. This seal controls the leakage of helium from the high pressure buffer to the low pressure buffer.

-The low pressure secondary seal could have been a ring seal since differential pressure across this seal is only 0.15 bars. A face seal was chosen to reduce seal development effort to a single type of seal.

- With 26 bars across the high pressure secondary face seal and 0.15 bars across both the primary labyrinth seal and the low pressure secondary seals, the following seal leakage rates are computed:

-1500 Nm3/hr buffer helium leakage through the primary labyrinth seal to the PC vessel.

-150 Nm3/hr buffer helium leakage through the high pressure secondary seal to the low pressure helium scavenge.

-2.5 Nm3/hr buffer helium leakage across low pressure secondary seal to the environment

-zero leakage of PC vessel helium to the environment

An alternate configuration could incorporate a high pressure air buffer replacing the low pressure helium cavity. This would require a large air compressor, but would significantly reduce the leakage of buffer helium at the expense of large air leakage to the environment. Assuming that a 0.15 bars pressure differential is maintained between the high pressure helium buffer and the high pressure air buffer, the total buffer helium loss would be 2.5 Nm3/hr.

Feasibility considerations beyond satisfaction of life and operating requirements include seal packaging and ease of installation and replacement.

The buffered dry seal configuration will require plumbing high pressure helium from the helium purification system and returning low pressure buffer helium back to the cycle.

Industry experience has shown that dry gas seals are a viable, cost saving replacement for oil lubricated ring seals operating in hydrogen and steam. Current dry gas seal designs operating in steam or hydrogen have a life expectancy of 5 years. Wear out is the primary life limiting characteristic of the dry gas face seal.

The large size and the required precision required for sealing applications also generates a considerable manufacturing challenge.

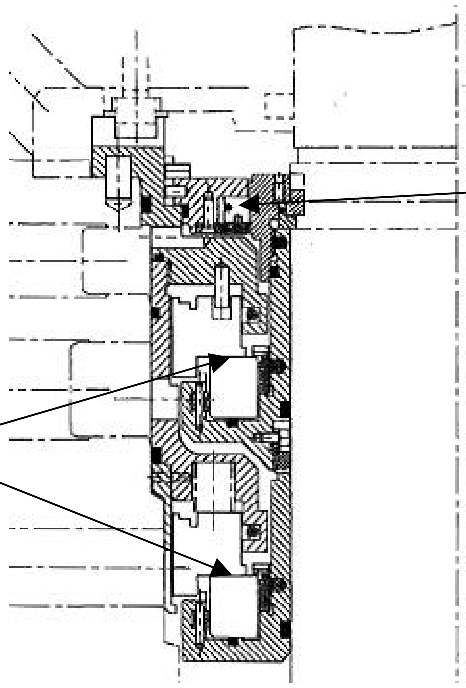
Seal design procedures will insure freedom from harmful resonances that coincide with turbomachinery excitation sources. The low magnitude and frequency of seismic excitation will not influence seal operation.

In summary, a buffered dry gas seal system appears capable of meeting leakage requirements, but requires a considerable extension of existing technology to produce the necessary life in a dry helium environment. Additionally, the precision required for the large size constitutes a major manufacturing challenge.

5. Example of dry gas seals

5.1 JEUMONT dry gas seal

Jeumont has developed a dry gas seal which is made up of three successive seals.

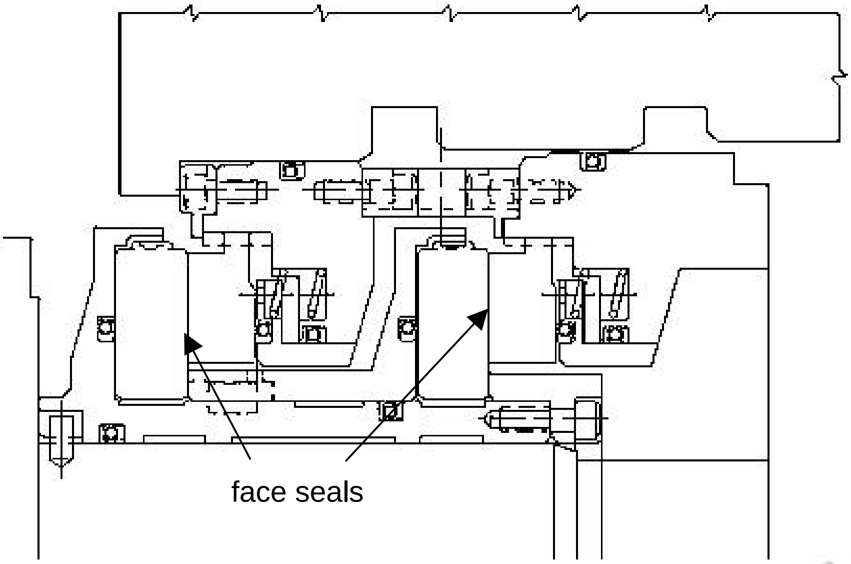


The two first seals are dry gas face seals: rotor faces rotate in front of floating elements faces which are balanced under 3 main forces: the static differential pressures acting on the element surface, the hydrodynamic pressure (partly due to Rayleigh blocks on the rotor surface) acting on the gap surfaces and the spring load. This various forces determines the magnitude of the film thickness. Element faces must remain flat whatever the service pressure of the seal. When the shaft stops turning, floating elements form an effective static seal against the rotor. The secondary seals required to prevent helium leakage downstream the floating element (stator) is provided with piston rings. A radial packing ring provides a third auxiliary seal. The operating life seems to limited due to wear incurred on the sealing surface.

The JEUMONT dry gas seal (for 102 mm diameter shaft) has been tested at a 14000 rpm speed with an inside pressure of 110 bars. The helium leakage rate measured is about 13 Nm3/hr. These test conditions don't fit with HTR vessel conditions. The shaft diameter is smaller (about 450 mm for GT-MHR shaft), the rotation speed is higher (3000 rpm for HTR turbine) and the inside pressure is higher (about 26 bars in the power conversion system vessel). Owing to the fact that the leakage rate depends on the shaft diameter, the film thickness (between 2.5 and 3.2 µm) and the inside pressure, tests have to be achieved with HTR parameters. It must be kept in mind that the large size and the precision required for HTR seal surfaces generates a manufacturing challenge. Besides, owing to the wear during transient speeds but transient pressures and temperatures too, this seal needs shutdown maintenance to replace faulty parts by seal elements in good working order.

5.2 JOHN CRANE dry gas seal

John crane dry gas seal is nearly the same as the JEUMONT system. Only two successive dry face seals are successively used.



Their operating conditions are:

- temperature: up to +315°C

- pressures: up to 200 bars g

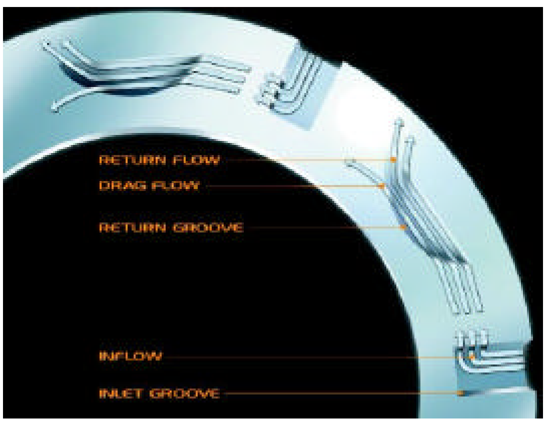
- linear speeds: up to 180 m/s (it corresponds to a rotation speed of more than 7000 rpm on a 450 mm diameter shaft)

- shaft sizes from 25.4 mm to 254 mm

Characteristics of the seal:

- during dynamic operation, a sealing gap of approximately 5 µm is maintained between faces, thereby eliminating wear.

- gas leakages typically controlled to about 3.15 Nm3/hr or less.

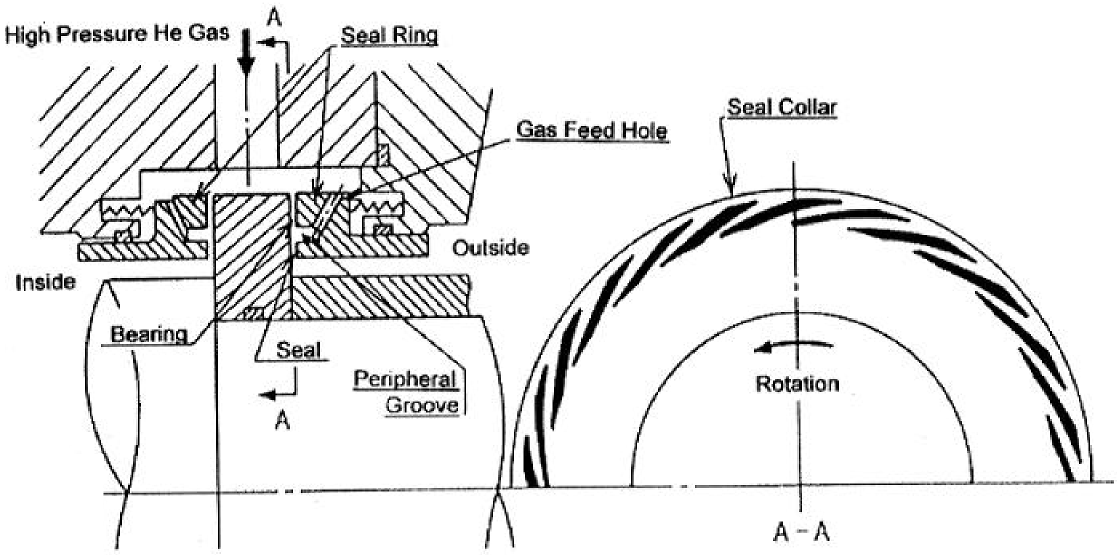


Hydrodynamic pads used by John crane for dry gas face seals

5.3 JAERI dry gas seal

The figure below shows a scheme of spiral groove face seal. The feature of this structure is to divide the rubbing surface into a seal section and a bearing section by providing circumferential groove on the seal. In the bearing section, spiral grooves are machined on the collar and the working gas is introduced from inner radius to the outer radius along the grooves with the rotation of seal collar. The seal ring has gas supply holes connecting the groove and outer peripheral gas plenum. Through these penetration holes, both the gas is supplied into the seal section and the bearing section.

This system is an other type of buffered system with dry gas face seals.



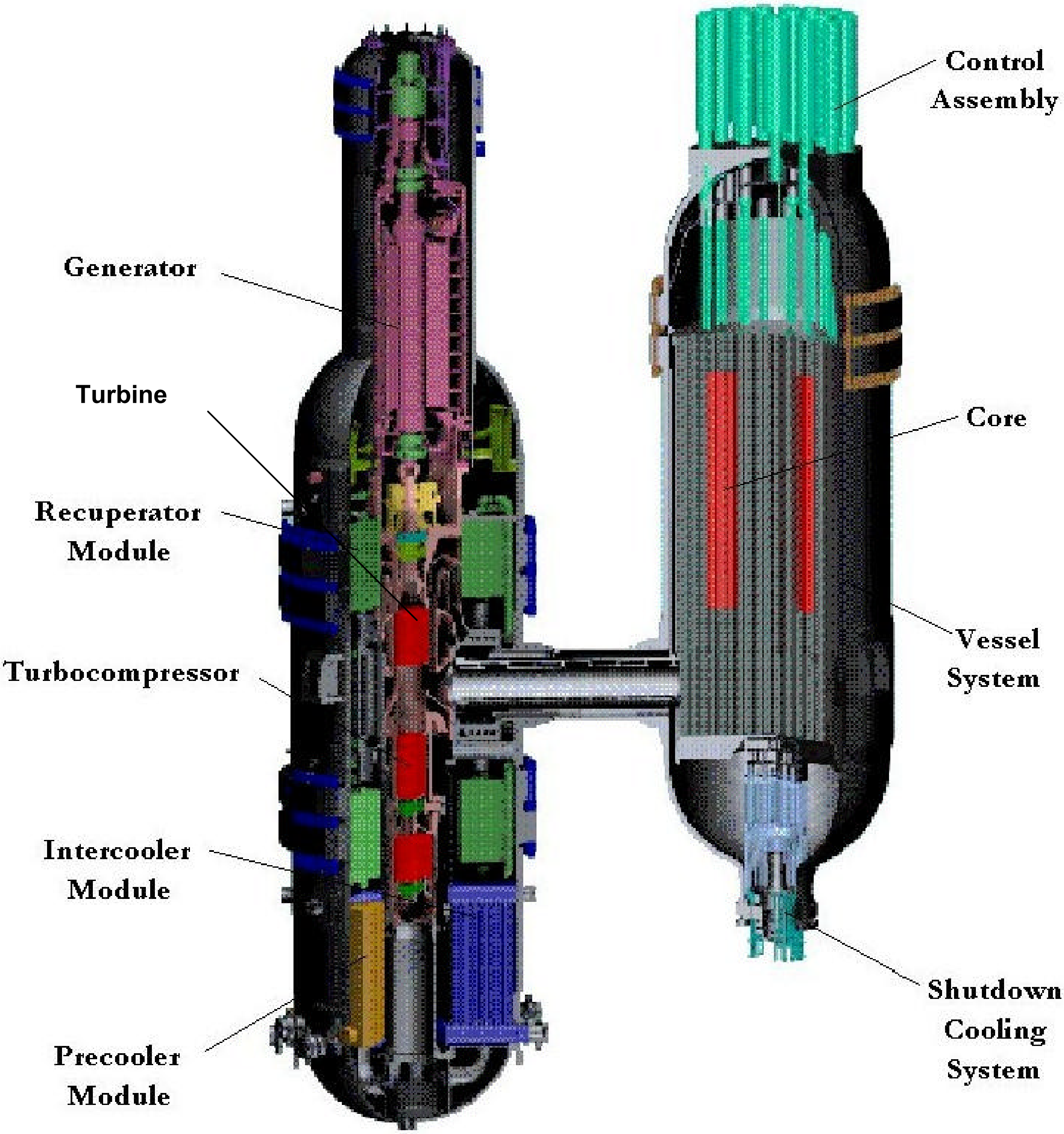
6. Conclusion

Dry gas performance has not been demonstrated in concurrent satisfaction of GT-MHR type or PBMR type requirements of size, temperature, pressure and dry helium environment.

It is apparent, as a first assessment, that helium leakage requirements cannot be satisfied with a single dry gas face ring or face seal. A buffered dry gas seal system appears capable of meeting leakage requirements but requires a considerable extension of existing technology to produce the necessary life in a dry helium environment. Additionally, the precision required for the large size constitutes a major manufacturing challenge.

Appendix 1

GT-MHR module arrangement



GT-MHR MODULE

Appendix 2

PBMR module arrangement

	Helium rotating seal	**EVEK DC 02 0250 /** Rev : A A_2 2/2