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SUMMARY AND CONCLUSIONS

Given its operating requirements, the rotating seal system represents a highly sensitive technical point in the HTR (High Temperature Reactor) design program.

The 450 mm diameter shaft is immersed in helium at the turbine end, at a temperature of 110 °C and under a pressure of 26 bar (in normal operating mode), and turns at a speed of 3,000 rpm (or 3,600 rpm).

Drawing on bibliographical research on rotating seal technologies for gas applications, this study has picked out two possible technical solutions, the first combining a labyrinth seal and mechanical seal, the second combining a labyrinth seal and brush seal. Other solutions, such as those based on ferro-fluid for example, are not ruled out, as they could be used in addition to the main systems.

The final choice of technology cannot be made until all operating conditions have been considered, which means not only nominal and exceptional conditions, but also those encountered during transient phases, as both these sealing technologies are affected when the direction of rotation is reversed (although mechanical seals can withstand this phenomenon for a limited time). Other aspects to be considered include heat removal, maintenance, and wear.

Notes:

(1) Security level (delete where inappropriate)

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APPRO	Procurement specification
CON	Inspection specification
CR	Report
DF	Supplier document
EQUIP	Equipment specification
FMO	Procedure data sheet
NT	Technical memorandum
PROC	Procedure
QMO	Procedure qualification
RECETTE	Acceptance specification
RES	Test report
RET	Design report
RT	Technical report
STAGE	Training course report

CONTENTS

1. INTRODUCTION

Given its operating requirements, the rotating seal system represents a highly sensitive technical point in the preliminary design work on the GT-MHR or PMBR models of the HTR (High Temperature Reactor).

As can be seen in the schematic diagram of the GT-MHR model in Appendix A, the 450 mm diameter shaft is immersed in helium at the turbine end, at a temperature of 110 °C and under a pressure of 26 bar, turning at a speed of 3,000 rpm (or 3,600 rpm).

This feasibility study has the following purposes:

- to examine the various techniques available for ensuring the tightness of this type of joint,
- to carry out bibliographical research with suppliers and users to select technologies capable of meeting technical requirements.

Alongside this study, Framatome ANP Jeumont is examining a device consisting of a dual system combining a labyrinth seal and a sealing gas barrier.

2. REFERENCE DOCUMENTS

[1] – Helium rotating seal. Feasibility study.
Jeumont Report HTR-E 03/07 D422.

[2] – Mechanical function and components. Engineering techniques
(Fonctions et composants mécaniques. - Techniques de l'ingénieur.)

[3] – Helium rotating seal. State of the art and specifications.
Framatome ANP Report HTR-E 02/06 D410.

3. TECHNICAL REQUIREMENTS

Normal or exceptional operating conditions are as follows:

- operating fluid: dry helium,
- operating pressure: 26 bar,
- operating temperature (near the leak): 110 °C,
- normal design temperature: 150 °C,
- exceptional temperature for a short period: 450 to 500 °C,
- operating rotation speed: 3,000 rpm (approx. linear speed 80 m/s),
- possible overspeed: 3,600 rpm,
- shaft diameter: 450 mm minimum,
- axial clearance: 0.5 + approx. 10 = 10.5 mm,
- radial clearance: 0.4 mm,

- seal lifetime: 60 years with inspection and regular replacement of parts (as infrequently as possible),
- required leak rate:
  - GT-MHR model: 10% by weight of primary helium/year, i.e. 0.31 Nm3/h (55 g/h),
  - PBMR model: 0.1% by weight of primary helium/day, i.e. 0.86 Nm3/h (154 g/h).

HTR prototypes, FSV (Fort Saint Vrain - USA) and HTR Steam Cycle in Germany, have shown that it is technically feasible to obtain a value of 0.1% a day, with a permissible maximum of 1% a day for FSV.

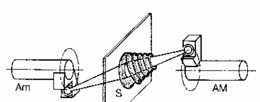
Leak rates are only considered for primary helium containing active impurities.

If the seal system includes a gas barrier with clean helium from an auxiliary system, it would be reasonable to aim for a leak rate equal to the volume of primary helium per year, i.e. 0.27% a day. This would also be more economical in terms of development costs. Likewise, a maximum permissible value of 1% a day is recommended for FSV.

Jeumont's preliminary design study (see ref.[1]) showed that it was technically possible to achieve a leak rate of 2.5 Nm3/h, i.e. 0.22% a day.

4. ROTATING SHAFT SEAL TECHNOLOGIES

This chapter is taken mainly from reference document [2].

<table>
<tr><th>Type</th><th>Operating principle</th><th>Diagram</th><th>Advantages
and drawbacks</th></tr>
<tr><td>By deformation (positive seal).</td><td>Deformation of a flexible part (metal bellows).
The most common types operate with a rotating oblique shaft with a swivel mid-point, or a cranked or eccentric shaft.</td><td>

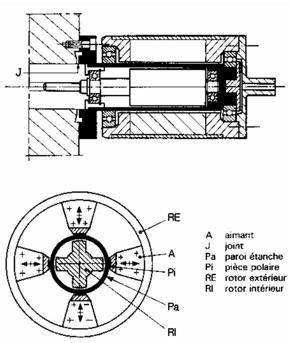
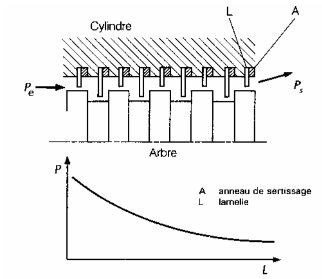
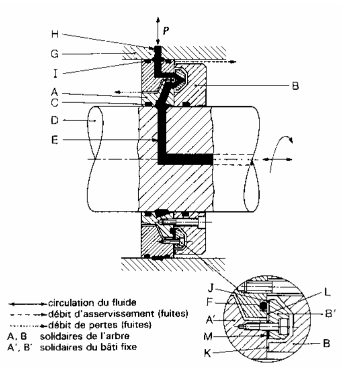
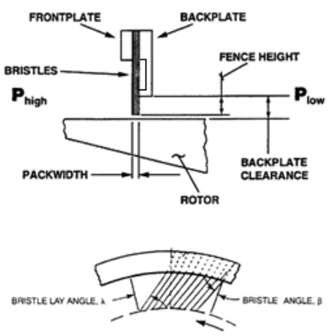
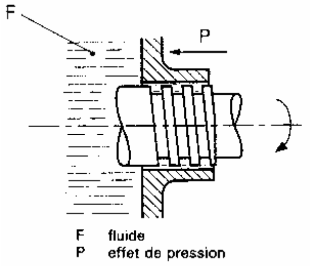
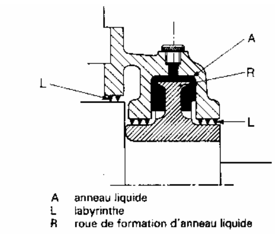
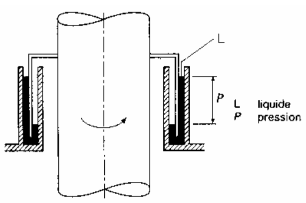
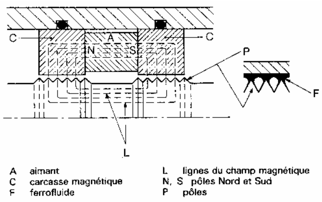
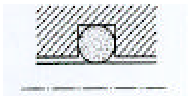
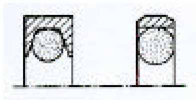
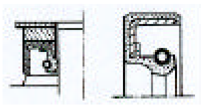
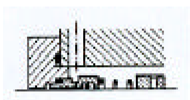
(a) avec arbre oblique
Am arbre mené
AM arbre menant
S soufflet</td><td>• Pressure up to 10 bar.
• Rotation speed of several hundred revolutions per minute.
• Complex and expensive.
• Not widely used, except for hazardous, radioactive fluids or ultra-high vacuum environments.</td></tr>
<tr><td>Magnetic transmission (positive seal).</td><td>A magnetic part integral with the drive shaft drives another magnetic part integral with the driven shaft. A non-magnetic, relatively fine, metal separator is inserted in the field air gap.</td><td>

Fig. 69. – Traversée magnétique étanche.</td><td>• Can handle high speeds, although there is a risk of stalling.
• Transmits torque values up to 100 N.m, with quite small volumes.</td></tr>
</table>

Labyrinth seals (controlled-leakage seal).	Makes the fluid flow over a longer leakage path, thereby reducing the flow rate.	 Fig. 71. – Joint labyrinthe.	• Minimum clearance between the moving and fixed parts. • Quite common. • Good dusttightness.
Transfer seals (controlled-leakage seal).	A and B are connected to the shaft via elastic seals C. A' and B' are connected to frame G via seals I. The pressurized fluid arrives at frame G via channel H, and flows through A' and B' along groove F, which arrives in the same plane as sealing slots J-L. Holes in part A ensure the contact between groove F and channels E of shaft D. Slots J-K-L are located in the same plane. The clearance required for the system to operate is provided by spacer M. The hydrostatic control mechanism consists of slots J-K, the high-pressure surface defined by groove F, and the intermediate-pressure area bounded by slots J-K.	 Fig. 75. – Joint de transfert (d'après doc. Viscotherm).	• For liquids (especially oil). • Power loss due to leaks and viscous friction torque does not exceed 3% of transmissible power.
Brush seals (controlled-leakage seal).	The seal consists of thousands of wire bristles with an approx. diameter of 0.1 mm. The bristles can be made of steel alloy, stainless steel, or a cobalt- or nickel-based superalloy.		• Shaft diameters can be up to 2.5 m. • The direction of rotation must not be reversed. • Good dusttightness. • Can be fitted in one piece or in segments. • Takes up radial or axial displacements. • Surface treatment may be required on the rotor.

Viscous effect (kinematic-effect relative seal)	Consists of a threaded shaft that rotates in the opposite direction to its thread. It turns inside a bore with very little clearance, applying a pressure to the fluid that can balance the pressure of this fluid.	 **Fig. 78. – Joint à viscosité.**	• Only effective for rotation. Must be paired up with a static seal. • Can be used with liquids displaying considerable viscosity.
Centrifugal effect (kinematic-effect relative seal).	A liquid is made to rotate. Due to the effect of centrifugal acceleration, maximum pressure will be built up inside the liquid at a certain distance from the axis of rotation.	 **Fig. 80. – Joint à anneau liquide simple centrifuge,**	• Only effective for rotation. Must be paired up with a static seal. • For gas applications. The liquid ring is water (or a heavy liquid such as mercury).
Liquid bath (kinematic-effect relative seal).	A cylindrical plate integral with the shaft is plunged into a liquid bath. If the liquid wets the sides thoroughly, and if the pressure of the vapor of this liquid is ignored, sealing can be considered positive.	 **Fig. 82. – Étanchéité par bain liquide.**	• Only effective for rotation. Must be paired up with a static seal. • For gas applications only. • For vertical shafts only. • Low pressure and rotation speed.
Magnetic liquid (kinematic-effect relative seal).	In this case, a liquid contains a suspension of very fine magnetic powder (ferrofluid). If a magnetic field crosses it, the fluid acquires a rigidity that counters pressure in the axial direction, but allows low-torque rotation.	 **Fig. 83. – Étanchéité par fluide magnétique.**	• Only effective for rotation. Must be paired up with a static seal. • Allows pressures in excess of 1 MPa, and rotation speeds of thousands of rpm. • Used especially for vacuum and cryogenic applications.

In addition to these various sealing techniques, radial seals (fitted directly on the shaft) or axial seals (parallel to the shaft axis) can be considered. One of the factors characterizing these seals, given in the table below taken from reference document [2], is the PV factor, expressed in MPa.m.s^-1 and representing the product of pressure times velocity.

Type of seal	Diagram	Materials	V max. (m/s)	P max. (MPa)	PV factor max.	T max. (°C)	Comments and applications
Toroidal seal (radial)		Elastomer	0.5	10	1	200	Lubrication required. Useful inclined arrangement. Compact. Direction of pressure not significant.
Composite seal (radial)		Elastomer Plastic (PTFE)	2	8	2	220	Very fine roughness required. Costs more than O-ring alone. Compact, low torque. Direction of pressure not significant.
Lip seal (radial)		Elastomer Steel	25	1	1.5	180	Lubrication required. Shaft plunge cut grinding. Inexpensive. High-velocity device, but low pressure. Gearbox output, axle shaft.
Braided seal (radial)		PTFE Graphite	20	10	10	600	Break-in period and tightening adjustment required. Various rotating pumps and aggressive fluids.
Ring seal (radial)		Carbon Hardfaced bronze	15	10	2.5	400	Rather expensive and often displays slight leakage. For pump shafts with relatively low friction.
V-seal (axial)		Elastomer	15	0.01	0.1	100	Useful for any type of shaft, hardened friction plate. Low-pressure bearing seal.
Mechanical seal (axial)		Carbon Steel Ceramic Carbide PTFE Elastomer	100	25	300	400	Rather expensive. Requires very good fitting and high-quality rotation. Lubrication essential. Widely used for high-performance equipment.

5. POSSIBLE TECHNOLOGIES

Of all the technologies mentioned above, only a few can be given serious consideration:

- labyrinth seals,
- brush seals,
- and mechanical seals, which are the only ones that can meet the PV factor of our application (approx. 200 Mpa.m.s^-1).

Some technologies of relative tightness by kinematic effect, as the centrifugal effect or liquid magnetic, seem attractive but require the addition of a static seal in the absence of rotation. They require, moreover, the addition of a liquid, which besides the problem of containment, would be contaminated dusts carrier.

For our application, a labyrinth/mechanical seal or labyrinth/brush seal tandem arrangement could meet requirements.

In both cases, the labyrinth seal on the high-pressure side does not require a fitting with minimum clearance, as it does not perform the main sealing function. A fitting composed of 6 to 8 teeth should suffice.

5.1. Labyrinth and mechanical seals

This type of seal is widely used, in centrifugal compressors for example. Several arrangements are possible, as shown in the two figures below taken from the Flowserve manufacturer's catalog.

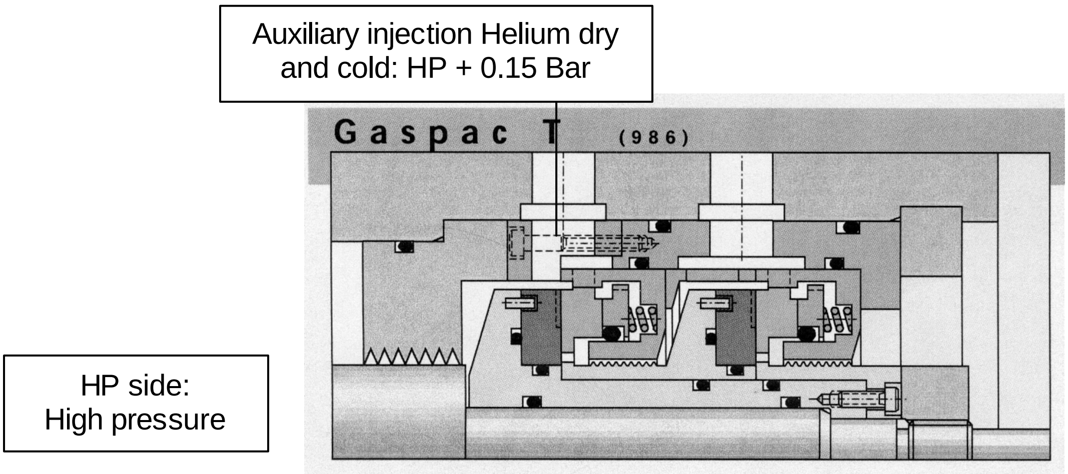
Both arrangements can withstand a pressure of 230 bar, a temperature of 230 °C, and a velocity of 200 m/s. The main problem here is the shaft diameter, which in our case is 450 mm, as opposed to the maximum diameter of about 300 mm recommended by manufacturers. According to the conclusions of document [1], using this arrangement for a 450 mm diameter would require considerable development work.

This type of arrangement could be improved by injecting barrier gas at a higher pressure than that of the process (in our case helium could be used). This would not only keep solid particles on the high-pressure side, but would also remove heat.

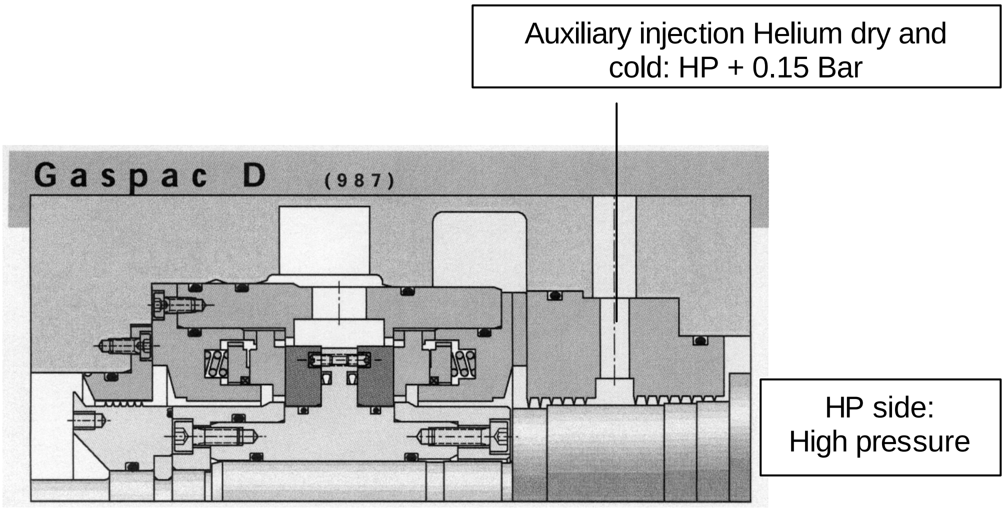
According to users, the shape of the grooves on the bearing faces (active interface allowing the leakage flow through and ensuring heat removal) is not significant. Seals composed of silicon nitride ring faces ensure good heat removal.

The second "face-to-face" arrangement is more difficult to cool and generally requires the use of cooling oil.

Cartridge seal tandem arrangement:



Cartridge seal "face-to-face" arrangement:



5.2. Labyrinth and brush seals

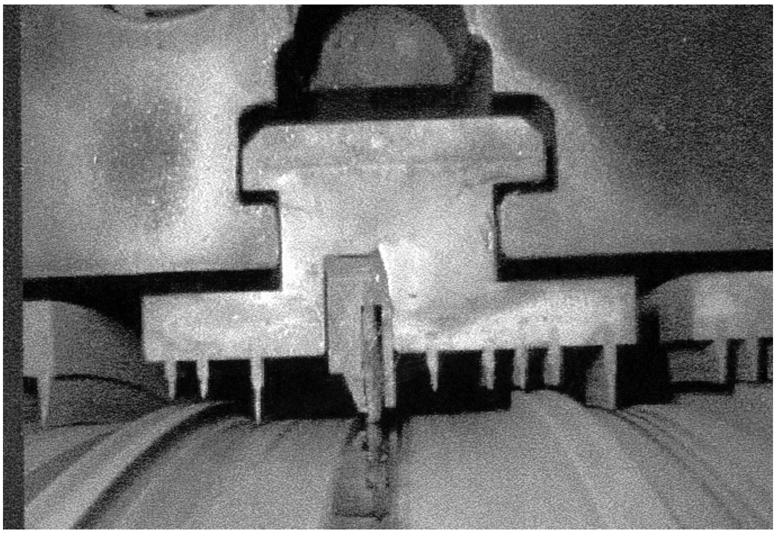
This type of arrangement is mainly used in the aerospace industry or on gas turbines. The example below is taken from a General Electric application.

The number of teeth in the labyrinth system can be reduced by adding a brush seal. Drawing on data from a NASA report (CR-135307), Technetics claims that if a brush stage is added to a labyrinth seal with 4 teeth and 0.51 mm clearance, leakage is 17 times less, and 8 times less for a clearance of 0.254 mm.

The brushes can be mounted on shaft diameters of up to 1,200 mm and at temperatures of 600 °C. The pressure drop of a brush stage can be 20 bar, and the velocity 384 m/s (data provided by Cross Manufacturing).

Like the labyrinth-mechanical seal arrangement described above, a barrier gas can be injected at a higher pressure than that of the process to keep solid particles on the high-pressure side.

Several brush stages can be mounted in series. In this case, the leak rate is expressed in $1/\sqrt{n}$ where n is the number of stages (e.g. with 2 brushes, the leak rate is reduced by 1.414).



Bibliographical research on brushes did not provide any information concerning wear in wires or the rotor part in a helium environment (the articles consulted, in particular NASA reports, are about aircraft engine tests and therefore concern propellant gases).

Tribological tests for selecting wire material (generally steel alloy, stainless steel, or nickel- or cobalt-based superalloys) and tests to determine whether any rotor coating is required should be performed to assess wear and maintenance requirements.

5.3. Comparison of techniques

	Labyrinth and mechanical seal	**Labyrinth and brush seal**
Sensitivity to shaft/bore concentricity problems.	Caution: mechanical seals must be perfectly aligned and, more importantly, rotating parts must rest perfectly on static surfaces. In our case, the shaft diameter makes this especially difficult (there is a risk of corner effect and overheating in some spots).	Theoretically, this should present fewer problems than the mechanical seal solution.

Shaft diameter 450 mm.	Manufacturers recommend a maximum diameter of approx. 300 mm.	This technique is used on shafts with diameters up to 1,200 mm.
Sensitivity to rotation reversal.	Caution: in most cases, cannot withstand reversal for more than 20 mn. Risk of shaft jamming and destruction of seal.	Impossible, the wires are inclined in the opposite direction and loss of sealing occurs.
Sensitivity to leakage flow reversal (LP side > HP side).	With the "face-to-face" arrangement, the seals may come permanently unstuck on shutdown, leading to a loss of sealing on restart (in spite of the bearing springs). In this case, dismantling for repair is necessary.	Theoretically, this has no impact.
Frictional wear.	Little impact.	Significant impact. The wire and shaft contact surface material pairs must be adapted (the shaft may require surface treatment).
Maintenance and spare parts.	The mechanical seal comes as a one-piece ring. In our case, this means dismantling the top.	The brush can be installed in several segments.

5.4. Additional systems

Given the performance levels expected from these 2 sealing techniques, with particular regard to the pressure drop on the LP side, an additional system (ferrofluid) could be added.

Taking the example of a secondary device with a clearance of 25 mm around the 450 mm shaft, and a pressure on the secondary side of 1 bar, the force to be applied to the helium leak is only 3.73 kN, instead of the 100 kN required for a pressure of 26 bar.

In the case of a vertical shaft, a liquid bath device (see section 4) could be used. The force could be taken up by the effect of a magnet on the particles in a ferrofluid bath and/or by the compression effect induced by a rotating ring immersed in the fluid bath.

Although this system offers the advantage of being perfectly leaktight, the literature mentions no applications for high velocities, such as the 3,000 rpm of our model.

6. CONCLUSIONS

This feasibility study concerning a rotating shaft seal for the HTR project turbogenerator, and drawing on bibliographical research into all existing technologies, shows the choice to be limited given the technical and environmental requirements.

Two options emerge – one for a labyrinth and mechanical seal, the other for a labyrinth and brush seal. These conclusions are in partial agreement with those given in reference document [3], which only recommended the first of these options. The brush seal option has, however, seen considerable development in recent years.

Given the performance characteristics of these two types of sealing system, another possibility that should not be ruled out is the addition of a secondary device (ferro-fluid or other system) downstream (LP side) to reduce, if not eliminate leakage.

Before a final decision can be made, a more thorough comparative study of the two technologies should be carried out, taking into consideration not only nominal and exceptional conditions, but also those encountered during transient phases (startup and shutdown), and in both static and dynamic terms. This is important because both sealing techniques are sensitive to a reversal in the direction of rotation, for example.

Other aspects not to be overlooked include heat removal, maintenance, and wear, with particular regard to material pairs.

APPENDIX A

GENERAL VIEW OF A GT-MHR MODEL AND LOCATION OF THE ROTATING SEAL SYSTEM