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Feasibility study

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HTR-E - WP4

HELIUM LEAKTIGHT ROTATING SEAL

FEASIBILITY STUDY (GT-MHR)

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Helium Leaktightness Rotating Seal

(2) FRAMATOME-ANP document : HTR-E 02/06-D-4-1-0
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(3) FRAMATOME-ANP (Eric Breuil) E-mail : dated February 18th 2003
Target and maximum values for a buffered dry gas rotating seal for GT-MHR and PBMR

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Dimensional data proposition for NRG calculations

1. INTRODUCTION

For a direct cycle HTR (GT-MHR type reactor – see Figure 1) considering a single shaft for all rotating components (turbo-compressor and generator), it could be interesting to locate the generator outside the pressure vessel. Then, the support design of the shaft could be more simple (located outside the pressure vessel, using a mechanical bearing, generator and turbo-compressor shaft separately supported…). Furthermore, as the accessibility to the generator becomes possible without opening the pressure vessel, the maintenance operations on this component are easier.

But this arrangement leads to insert a helium leaktight rotating seal on the primary helium containment to allow a passage for the rotating shaft.

The main objective of Work Package 4 is to identify the best promising design for such a rotating seal.

The tasks and deliverables of this Work Package are described in the reference (1) workplan document.

The first mentioned deliverable D30 (reference (2)) of the workplan presents the state-of-the-art of dry gas systems and defines the requirements for the rotating seal.

This document is the following deliverable D32 and is a feasibility study of dry gas system for the GT-MHR concept, as regards the HTR conditions. It analyses the existing devices, defines a suitable concept of dry system, and justifies it by theoretical analysis.

2. REQUIREMENTS

As mentioned in the introduction, the requirements for the dry gas system are defined in document reference (2). They are summarized in Table 1.

The requirements can be decomposed in three sets of data corresponding to :

- the operating conditions :

  The seal must restrict the leakage of helium at a pressure of 26 bars and a temperature of 110°C.

Nevertheless, the temperature to consider for the seal mechanical design is identical to the design temperature of the vessel, that means 150°C.
Furthermore, the seal must be able to operate during a short time at an emergency mode temperature between 450°C and 500°C.

The rotation speed of the rotor is 3000 rpm with a possible trip speed of 3600 rpm. On the other hand, the seal must be able to operate when the rotor is at rest (null rotation speed).

- the geometrical requirements :

The seal is mounted on a shaft whose diameter is higher or equal than 450 mm. This induces a nominal linear sliding speed between the sealing faces higher than 70 m/s, which can rise over 85 m/s in case of overspeed.

A shaft runout of 0,5 mm and a 0,4 mm radial rotor displacement may occur when the magnetic levitation is lost.

And the shaft thermal growth between the seal and the catcher bearing may result in a relative axial displacement of the seal of about 10 mm.
So the total relative axial displacement imposed to the seal may reach 10 + 0,5 mm = 10,5 mm.

- the seal maintenance requirements :

The installation or removal of the seal must be easy and of short time duration.

The possibility of testing before and during the installation must be taken into account.

- the expected seal performances :

A seal life of 60 year with periodic inspections and parts replacements, is aimed. The interval between the inspections / parts replacements must be as long as possible.

The seal leakage must be minimized. A leakoff flow rate of 10% of the helium mass corresponding to 0,31 Nm^3/h (helium at 1 atm, 0°C) is considered as a primary assumption.

3. BIBLIOGRAPHY – DRY GAS SEAL PERFORMANCES

A bibliographical examination of existing rotating gas seal characteristics is a good preliminary approach to get information about the required seal.

This examination has been performed with available data from well-known gas seal manufacturer such as John Crane, Flowserve, Kaydon, Sealol, Stein…

The maximum characteristics obtained for that kind of seal are :

- pressure : 230 bars
- temperature : 430°C
- rotation speed : 16000 rpm
- sliding linear speed : 200 m/s
- shaft size : 380 mm

Of course, each above mentioned value is considered independent from the other ones, because a seal operating in all these cumulated conditions would be too solicited, and is consequently not feasible.

These values gives a good indication the influence of each parameter on the seal feasibility.

It appears clearly that the required shaft diameter for GT-MH-R seal is higher than the maximum value noticed during the bibliographic examination. Most of the gas seal are much smaller with a typical diameter size between 100 mm and 200 mm. They often operates at high speed, which explains the high maximum linear speed above mentioned.

A high shaft size diameter induces large seals parts. The utilization of special materials (ceramics, graphite, elastomer) for the critical seal parts of these dimensions is a first difficulty. Furthermore the seal performance must be carefully predicted in order to choose the appropriated concept and design, and to identify the associated necessary systems (seal injection device…). This point and especially the seal interface behavior will be discussed later in the document.

On the other hand, the pressure, the rotation speed, the sliding linear speed, and temperature to a lesser extent (because of the high emergency temperature) should not be the main difficulties for the required seal, because all the corresponding required values are lower than the above mentioned values.

Nevertheless, as each of these parameters is considered independent from each other ones, the feasibility of a seal cumulating all these operating conditions is still to be evaluated.

Appendices 1 and 2 present two examples of dry gas seals tested by JEUMONT SA and FRAMATOME, and illustrate realistic performance expectable for dry gas seals.

They show that the pressure, temperature and linear speed should not be a real difficulty for HTR GT-MHR seal. On the other hand, the measured leakage flow rates are significantly higher than the required leakage. This point will be discuss later in the document with the interface behavior study.

4. DRY GAS SEAL CONCEPTS

Rotating shaft seals can be classified into the fixed clearance type and the surface guided type.

4.1 Fixed clearance seals

They are designed to never touch throughout the life of the seal (except during wear in or shaft excursion which in turn wear off the seal).

While there is a wide variety of types of fixed clearance seals, here are listed the most common types.

4.1.1 Bushing seal

This concept is illustrated by Figure 2. The flow area is the annulus created between the bushing and the shaft. Resistance to flow is determined by the length and the clearance. The clearance must be large enough to allow for all shaft motion.

4.1.2 Labyrinth seals

This is illustrated by Figure 3. This type of seal relies on creating a high head loss due to velocity loss in the cavities between teeth, to minimize leakage. This concept is presented and discussed in document (2).

4.2 Surface guided seals

They are those where one of the seal faces is flexibly mounted with respect to the shaft or housing and is thus entirely supported and guided by the second seal face, one of which is sliding relative to the other. Thus a surface-guided seal has certain characteristics common with a bearing. The leakage gap is determined by the character of the surfaces and the seal design. In some cases, the surfaces contact to at least some extent (they have no lift-off features) and would also be termed "contacting seals". In other cases, features such as lift pads, radial taper, spiral groove cause these faces to lift during operation such that they become "non-contacting". However in both cases, under static conditions, under zero pressure, all surface-guided seals contact.

These seals may be classified as to the type of guiding surface. These concepts are also described and discussed in document reference (2), and are here mentioned for reminder purpose only.

4.2.1 Ring seals

This concept is illustrated on Figure 4.

The dynamic sealing between the rotating shaft and the static housing is called the primary sealing and corresponds to the axial cylindrical area between a ring (generally in carbon) and the shaft. The initial clearance is limited by manufacturing capabilities and is in the range of 0.02 mm.

The ring is prevented from rotating for instance by means of anti-rotation slots as represented on the figure.

A secondary sealing is assured by the axial contact between the ring and its housing. A spring preloads the ring against the secondary sealing surface. In normal operating conditions, pressure distribution around the ring results in additional contact force at the secondary sealing surface.

Grooves or pads in the ring internal surface induce hydrodynamic forces that help the ring centering around the shaft.

This concept allows the required axial and radial displacements between rotor and stator.

On the other hand, the differential pressure is limited by the contact pressure at the secondary sealing surface and the corresponding induced wear.

Furthermore, the leakoff flow rate can not be reduced under a minimum value corresponding to the minimum possible clearance between the ring and the shaft. The possible contact between ring and shaft may also wear the sliding surface and increase the leakage.

And the thermal differential dilatation between the ring and the rotor may result in clearance and leakoff flow rate variation.

4.2.2 Face seals

This concept is illustrated on Figure 5.

The dynamic / primary sealing is radial and is located between a free to move axially ring called seal ring and its counter-face ring (called seal runner in the case of the figure because it is the rotating ring).

The seal ring is fixed in a support. The assembly (called the seal ring assembly) is free to move axially and is prevented from rotating by means of slots (as represented on the figure) or by means of a pin.

Springs between the seal ring support and its housing preload the seal ring on the seal runner.

In the present description the seal ring is not rotating, and the seal is called "stationary". In some other cases, the seal ring is rotating with the shaft, and the seal is then called "rotating".

"Rotating seal" present two main disadvantages :

- the seal ring self-positioning may be disturbed by the rotation with the shaft,
- the counter-face is sensitive, for high pressure to deformations of the housing on which it is fixed.

So, we will only focus on "stationary seals" for the present study.

The main secondary sealing is located between the seal ring support and the static internal housing in which it is located. This sealing is rather "dynamic" because it allows possible displacement of the seal ring assembly on its sliding surface.

All the other secondary sealings are totally static and complete the tightness. Concepts and materials of secondary sealings are discussed in §5.5.

Materials of seal faceplates (seal runner and seal ring) are discussed in §5.4.

Grooves or pads on the sealing face of the seal ring or the seal runner induce hydrodynamic forces that lift the seal ring and avoid or limit sealing faces wear.

Grooves design and influence is discussed in §5.2.2.

This concept allows the required axial and radial displacements between rotor and stator.

The operating differential pressure may be far higher than the required value mentioned in Table 1.

As the minimum gap between sealing faces is only limited by sealing faces flatness, the leakoff flow rate may be lower than the leakoff flow rate of seal ring concepts.

On the other hand this design is more complex and requires a careful design.

4.3 Concepts ranking

As a result as the previous description, the concepts may be ranked from the best promising to the less promising as follows :

1) Face seal
2) Ring seal
3) Fixed clearance seals (Labyrinth…)

5. SEAL DESIGN

According to the previous ranking, the following seal design considerations are developed in the case of a face seal concept.

5.1 Seal runner and ring loads

The seal runner is submitted to a :

- a lifting (i.e. opening) force due to high pressure under the seal runner,
- a seating (i.e. closing) force due to the interface pressure distribution,
- a possible additional seating force due to contact between the sealing faces.

The balance force applies the seal runner on its support on the shaft.

The seal runner is the guiding ring for the seal ring which is in equilibrium under the following loads :

- a seating force due to high pressure over the seal ring support,
- the weight of seal ring assembly,
- a additional seating force due to springs over the seal ring support,
- a lifting force due to the interface pressure distribution,
- a possible additional lifting force due to possible contact between the sealing faces,
- a rubbing force corresponding to contact between the secondary sealing and its sliding face. Its axial direction depends on the relative displacement between the seal ring and its internal housing.

When the interface fluid average pressure is greater than the sealed pressure, the seal is called unbalanced. This is usually the case of low pressure seals because of limitation due to interface heat dissipation.

When the interface fluid average pressure is lower than the sealed pressure, the seal is called balanced. This may be obtained with a suitable choice of balance diameter (see “Dbal” on Figure 5). The seal may then operate up to a far higher differential pressure, as already noted in §4.2.2.

A balanced seal is required for the present application.

5.2 Interface tribology

The interface pressure distribution and consequently the leakoff flow rate depends on the tribology concept.

5.2.1 Hydrostatic seal

Hydrostatic seal creates a balanced opening or closing force to maintain the just right amount of sealing faces separations. There is no contact between sealing faces.

Unlike hydrodynamic seals, the interface pressure distribution does not depend on the relative displacement due to rotation between the sealing faces. Two main concepts may be exposed :

- The hydrostatic radial taper seal

  The sealing faces are slightly slopped so that they form a convergent (open towards outside diameter and high pressure). This shape induces a pressure distribution and a lifting force depending on the film thickness between sealing faces. The force increases when the gap decreases, and decreases when the gap increases.

  The slopes must be very precisely machined and their deformation under pressure and temperature must be mastered to assure a good seal behavior.
  A minimum film thickness must be maintained to avoid contact between sealing faces and slopes modification because a small interface profile degradation results in a leakoff flow rate increase over the acceptable limits.
  Moreover, a minimum differential pressure is required to establish the minimum gap necessary for dynamic (i.e. rotation) operation of the seal.

  This technology is quite applicable for liquid seal, and it is mastered by Jeumont who manufactures high pressure rotating seals for primary pumps for pressurized water reactor.

  On the other hand, it is not used in practice for gas seals, and therefore lacks of field experience information. It is here mentioned for theoretical information.

- Other hydrostatic seals.

  The hydrostatic effect may be obtained by other means. For instance an orifice may be used as the controlling element. The principle is illustrated by Figure 6.

Pressurized gas comes in behind the seal ring assembly in the region bounded by the two secondary seals. These orifice cascades cause a flow dependant pressure drop between the pressurized gas and the faces. As the film thickness decreases and leakage decreases, the pressure drop across the orifice decreases and the average faces pressure increases. When the seal gap opens, a pressure drop across the orifice causes the face pressure to become smaller. By balancing the seal carefully, it will obtain an equilibrium value of film thickness that gives complete separation.

Unlike radial taper hydrostatic seal, the sealing faces are not slopped but flat. This allows occasional contact between the sealing faces if the faceplates materials are correctly chosen.

5.2.2 Hydrodynamic seal

The interface pressure distribution results of relative displacement between sliding sealing faces and suitable interface shape in the circonferential direction.

From a theoretical point of view, a wavy sealing face (as represented on Figure 7) may induce hydrodynamic effect as a result as the “oil corner” effect.

In practice, grooves are machined on one sealing faces (seal ring or seal runner) to increase significantly hydrodynamic effect and lifting force (see example on Figure 8).

This is the phenomenon you may experience when you trap water in the thread of your car tyre causing the car to hydroplane and lift off the road surface.

Unlike liquids, gases are compressible but do generate similar lifting force and good film stiffness if the seal geometry is designed and built correctly. The idea is to direct the gas into some narrow channels that increase the gas pressure causing the face separation.

The spiral grooves represented on Figure 8 have a pumping effect against or with the pressure gradient depending of the rotation direction. The pressure gradient is created in the grooves, and induces an additional lifting force.

Most designs are unidirectional and may operate in only one rotation direction. Nevertheless, some other designs such as the one represented on Figure 9 are symmetric / bi-directional and allow rotation in both directions.

Seals manufacturers have defined a lot of groove designs corresponding to their own know-how. Figure 10 shows some examples of these. Each of them requires an experimental or theoretical parametrical study to evaluate / determine their efficiency.

No literature or test data seems to exist and demonstrate a real superiority of a specific groove design.

Raleigh step pads is an other design (see Figure 11) which has been theoretically and experimentally evaluated as a hydrodynamic gas seal. It has already been studied and tested by Jeumont in the application mentioned in §3 and described in Appendix 1. It is the most commonly used technology together with spiral grooves.

In reality, these hydrodynamic seals induce both hydrodynamic and hydrostatic effects because the sealing interface is divided in two areas (see Figure 11).

The grooved area generates the hydrodynamic effect. It may also contribute to a small portion of the hydrostatic lifting force.

The sealing area (also called DAM and located downstream the grooves in the interface) is only submitted to a differential pressure and has a hydrostatic operation. Its pressure distribution determines the seal leakoff flow rate.

Film thickness of hydrodynamic gas seals is usually about three to four microns. This limits leakoff flow rate to lower values than those corresponding to hydrostatic seals.

On the other hand, hydrodynamic grooved seals may be sensitive to particles, and grooves depth modification (because of particles or by rubbing wear) must be avoided to maintain the required hydrodynamic effect.

5.3 Seal concept and interface for HTR-E – GT-MHR

Hydrostatic seal technology is not very often used alone for dry gas seal. Most of gas seals use some type of hydrodynamic lift pads in combination with hydrostatic feature to assure adequate lift-off and film stiffness.

The feasibility study will be carried on a seal using this technology association.

As no literature or test data seems to exist and demonstrate a real superiority of a specific groove design, the study will consider Raleigh step pads one sealing face because it is together with the spiral grooves the most common used grooves shape. Furthermore Jeumont has a previous satisfactory experience with such a design (see Appendix 1) and its geometrical shapes allow semi-manual preliminary sizing calculations.

Nevertheless, the hydrostatic technology may be considered as an alternative and emergency design solution in case of further difficulties with the chosen concept. Among the two possibilities presented in §5.2.1, the hydrostatic seal with an orifice controlling element seems to be more promising for gas seal because its flat sealing faces should be less sensitive to possible contact.

5.4 Sealing faces materials

For an hydrodynamic seal, the sealing faces material must have good sliding properties and the required temperature and pressure resistance.

In most applications, a combination carbon graphite / silicon carbide ceramic is used with satisfaction. These materials are suitable and then proposed for the HTR-E dry gas rotating seals assemblies.

The silicon carbide is recommended for the seal runner because its higher mechanical strength is more indicated for the rotating faceplate. Moreover its higher thermal conductivity allows a good distribution of the heat generated by the rotation limiting the sealing face deformation. The manufacturing of a silicon carbide ring of such a high size may be a concern. Nevertheless, it is feasible because similar rings have already been manufactured for a Jeumont seal a few years ago.

The carbon graphite is the seal ring material. As the limited level of oxygen in helium may impact graphite rubbing properties, a suitable nuance of graphite (impregnation with special additives) will have to be defined and evaluated. In case of problem (unacceptable additive in graphite nuance, or insufficient rubbing properties), an alternative material will have to be defined and estimated.

5.5 Secondary sealing

As explained in §4.2.2, the main secondary sealing is "dynamic" because the seal ring assembly moves on its sliding surface of the "internal housing". All the other secondary sealings are static (no relative displacement between parts).

In most of similar applications, elastomer (ethylene propylene for instance) O-rings is the standard solution used. This is suitable here during normal operation at temperature lower or equal to 150°C.

Unfortunately, the emergency situation with a temperature between 450 and 500°C do not authorize this material solution.

This requires the utilization of metallic seal (Helicoflex® for instance) for the static sealings. Some metallic O-rings are recommended to replace elastomer seals when these can not be used.

A piston ring may be used for the "dynamic" secondary sealing. This solution has a high temperature resistance. On the other hand, it induces a supplementary leakage and a possible wear of the sliding surfaces limiting the seal life duration. This will require a particular attention for the material choice of the sliding surfaces.

5.6 Preliminary interface calculations

A preliminary evaluation of the leakoff flow rate may be carried out with the following assumptions :

- Laminar flow
- Perfect gas
- Neglect effect of radius on flow area (i.e. valid for narrow seals)
- Isothermal conditions
- Subsonic flow

These assumptions are appropriate to compressible flow when the film thickness is very small and flow rate is not large. This means the flow is laminar and subsonic. The isothermal assumption is good for thin film of flow between two isothermal bodies so that heat is transferred to the gas. The perfect gas and neglect of the radius effect are made so as to be able to derive a relatively simple mathematical equation.

Leakoff flow rate is expressed versus interface gap height when sealing faces are flat and parallel, by the following equation (hydrostatic situation):

$$Q = \frac{\Pi \times D \times T_{std} \times h^3}{24 \times \mu \times T \times ?r \times P_{std}} \times \left(HP^2 - LP^2 \right)$$

with :

h = Interface Gap height

Δr = Sealing surface (DAM) width

μ = Dynamic viscosity

HP = High pressure

LP = Low pressure

D = Average sealing surface diameter

P_{std} = Reference pressure

T = Temperature

T_{std} = Reference temperature

The formula shows that the most important parameter is the film thickness. The aim is to have a minimum gap height to limit the leakoff flow rate, but high enough to limit possible contact between sealing faces.

It also indicates that increasing the sealing surface width is another useful way to limit the leakage.

For a leakoff flow rate = 0,31 Nm^3/h, and the dimensions range corresponding to Table 1 the required gap height is close to 2 µm.

This gap height is close to or lower than the usual value mentioned in the literature for gas seal with small diameters.

It is close to the maximum value of sealing faces flatness. This could lead to :

- a real gap higher than required which will result in a too high leakage,
- contact between the sealing faces and then possible wear leading to high leakage.

For the high diameter of the present project, the gap height should then be at least several microns higher than the required gap height of 2 µm, which would therefore result in higher leakoff flow rate.

So the target leakage flow rate for GT-MHR (0,31 Nm^3/h) cannot be obtained with a single seal, and requires a seal assembly. This solution is discussed in §6.

5.7 Further comments

This paragraph lists a few other important points to keep in mind for further design step.

- Sealing face deformation
  Because of the GT-MHR large shaft diameter, sealing faces deformations (after manufacturing or during operation) will be higher than for usual standard gas seals (diameter around 100 o 200 mm). This will require a sufficient film thickness to avoid unexpected contact sealing faces.

- Contact
  As specified in §5.4, sealing faces materials of gas seal should have good sliding properties. Grooves are often placed on the carbon-graphite seal ring sealing face. If unfortunately a wear by sliding contact occurs, the grooves depth may decrease

and modify the lifting force and the hydrodynamic effect. This could be avoided, if necessary, by placing the grooves on the harder sealing faces (the silicon carbide runner for instance).

- Groove clogging
  Groove clogging should be prevented to limit seal sensitivity to particles.
  This requires a fine filtration of seal gas before seal entrance to avoid particles arrival at seal inlet.
  Particles may be generated by rubbing contact between sealing faces. A high operating gap is another suitable way to limit the possibility of grooves clogging.

- Transient operating conditions
  Temperature and pressure transients induce interface profile modification which may temporary open or close. After a sufficient operation the sealing faces will be lapped and become flat and parallel.
  This phenomena will have to be minimized to assure the longest seal life duration.
  This will require to :
  - Limit the transient operations (amplitude and occurrences number)
  - Design the seal shape in order to minimize seal face deformation during transients.

- Vibrations influence
  They impact seal performance and life duration. Pump design will have to include this factor.

6. DRY GAS SEAL ASSEMBLY CONCEPTS

6.1 Assembly concepts discussion

This target leakoff flow rate of 0,31 Nm^3/h value corresponds to helium with impurities activity. In case of clean helium, the acceptable flow rate could be higher. Document reference (3) proposes a target value of 2,5 Nm^3/h and a maximum admissible value close to 11 Nm^3/h.

This can be performed with the combination of a barrier of clean helium injection to prevent leakage with impurities activity, and a seal assembly to limit the leakage value.

There are three main types of dual seal arrangements illustrated by Figure 12 :

- Back to back (seal rings are facing each other)
- Face to face (seal runners are facing each other)
- Tandem (first seal ring is facing second seal runner)

"Back to back" and "Face to face" arrangements require gas injection, at a pressure higher than pressure vessel, between the two seals. They are the less suitable solutions because outside leakoff flow rate is not salvaged.

Tandem arrangement with a clean helium injection upstream the first seal (vessel side) is better adapted because the pressure drop is divided between the two seals, which allows to reduce the outside leakoff flow rate.

Figure 13 represents the proposed assembly for this work package. It is composed of a tandem arrangement of two seals associated to a labyrinth with a clean helium injection between the first seal and the labyrinth.

The two seals are identical in order to limit development costs. They are in a cartridge assembly for an easy installation / removal on / of the PCS in order to reduce maintenance duration and operators irradiation doses.

The injection pressure is about 0,2 bar higher than the vessel pressure. This creates the barrier flow through the labyrinth. This barrier prevents helium with impurities activity to leak outside the vessel, and limits the seal operating temperature.

The pressure between the two seals is 0,2 bar higher than the outside (and atmospheric) pressure. So the first seal is submitted to the full pressure drop, and its leakage is clean and collected in the leakoff line and recycled. And the second seal operates under a 0,2 bar differential pressure with consequently a very small leakage lower than the target value (2,5 Nm3/h) of clean helium outside the vessel.

If the injection is lost, the vessel helium is leaking. The N°1 seal inlet pressure is only reduced by 0,2 bar and most of the leakoff is still collected in the leakoff line. The N°2 seal differential pressure is unchanged, and the leakoff flow rate outside the vessel is maintained to a value lower than 2,5 Nm^3/h.

In case of N°1 seals failure, the leakoff line between the two seal is closed and the N°2 seal withstand without damage the full pressure drop. As the injection is normally available (a simultaneous N°1 seal failure and loss of injection is very improbable) only clean helium is leaking outside the vessel and the N°2 seal limits the leakoff flowrate to a low value. Furthermore, if the rotation is stopped, the hydrodynamic effect disappears, which reduces the lifting force on the N°2 seal ring and the leakoff flow rate.

Some variant of the arrangement of Figure 13 may be defined.

For instance, if the flow rate into the vessel is considered too high, the labyrinth may be replaced by a third seal (a ring or face seal mounted face to face with the N°1 seal).

If the third seal is a ring seal, the N°2 seal may also replaced by an identical seal ring to limit the costs.

6.2 Proposition for further calculations

This paragraph presents the proposed outlet data for the "Stability Leakage Analysis" corresponding to task 1.2.4 of the reference (1) workplan document.

Dimensional and constructive data have been transmitted (document reference (4)) to NRG for further calculations. They correspond to :

- interface dimensions mentioned on Figure 15,
- springs and weight force,
- balance diameter "Dbal" (see Figure 13).

The grooves are located on seal ring sealing face, because their machining is easier on carbon graphite than on silicon carbide.

The grooves could also be machined on the seal runner (see §5.7 2nd bullet). This requires a small shape modification of the seal ring and the seal runner as represented on Figure 14. In that configuration, Figure 15 notations would refer to seal runner instead of the seal ring.

Spiral grooves could be also evaluated as an alternative to the proposed Raleigh pads.

Two hypothesis on the balance diameter "Dbal" have been transmitted :

For the N°1 seal, the 1^{rst} hypothesis should lead to lower seating force and then to higher gap and leakage flowrate than the 2nd one. It is more recommended for the sealing faces solicitations. On the other hand, the 2nd hypothesis is better on a leakage point of view.

The two hypothesis shouldn't lead to significant results differences for N°2 seal because of the very low differential pressure of that seal.

7. CONCLUSION

The required values of pressure, temperature and linear sliding speed are acceptable for the dry gas seal system of the Work Package 4 of HTR-E project.

The target leakoff flow rate can not be obtained with a single seal and requires a seal assembly.

The proposed solution is composed of a tandem of two hydrodynamic seals with Raleigh pads, a labyrinth between the vessel and the first seal, and an injection of clean helium between the labyrinth and the first seal.

It allows to meet the leakage requirements, and increases operating security margins in some emergency situations such as the failure of the first seal.

Dimensions and data sets corresponding to this solution are proposed for the "Stability Leakage Analysis" corresponding to task 1.2.4 of the reference (1) workplan document.

The large shaft size is the challenge of this work package part, because of the lack of field experience with very high size gas seals (deformation of sealing faces, vibration influence…).

Hydrostatic seal concept with an orifice controlling element is another alternative and promising solution which could be to evaluate if necessary.

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

Table 1 : Requirements for the dry gas system

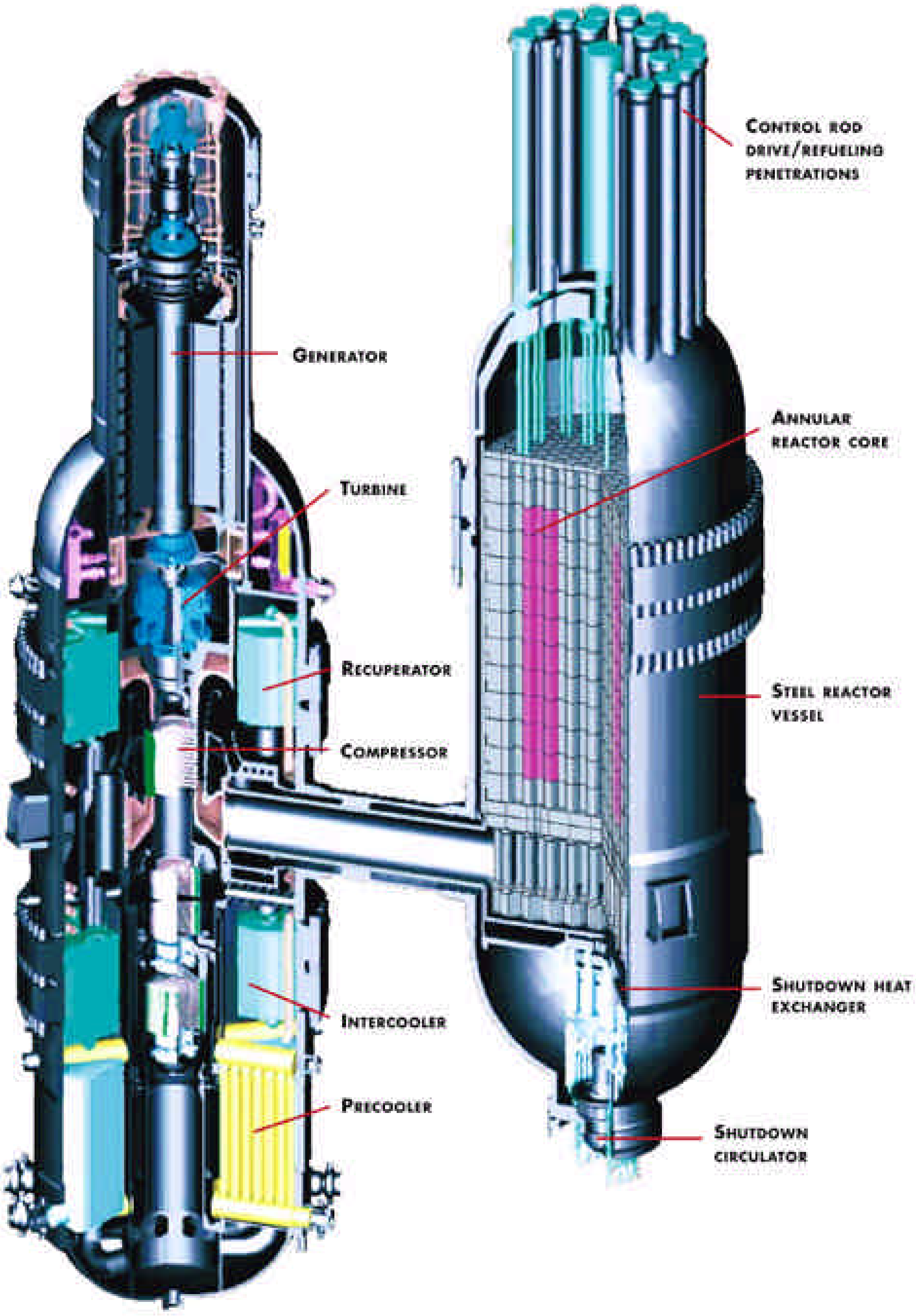


Figure 1 : GT-MHR module

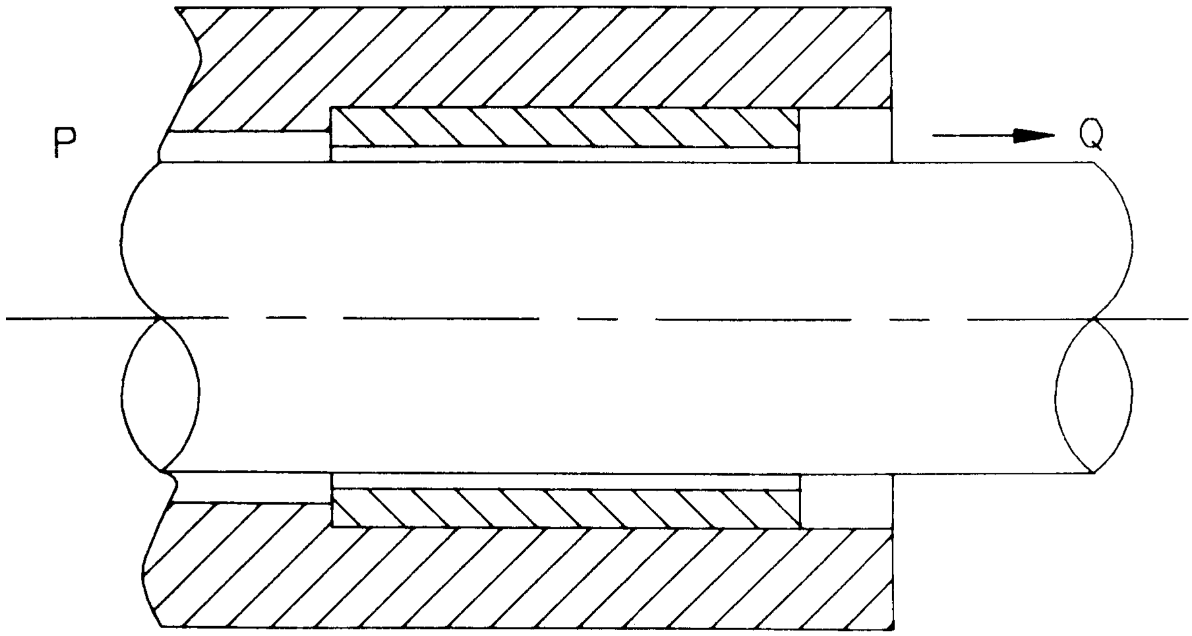


Figure 2 : Bushing seal

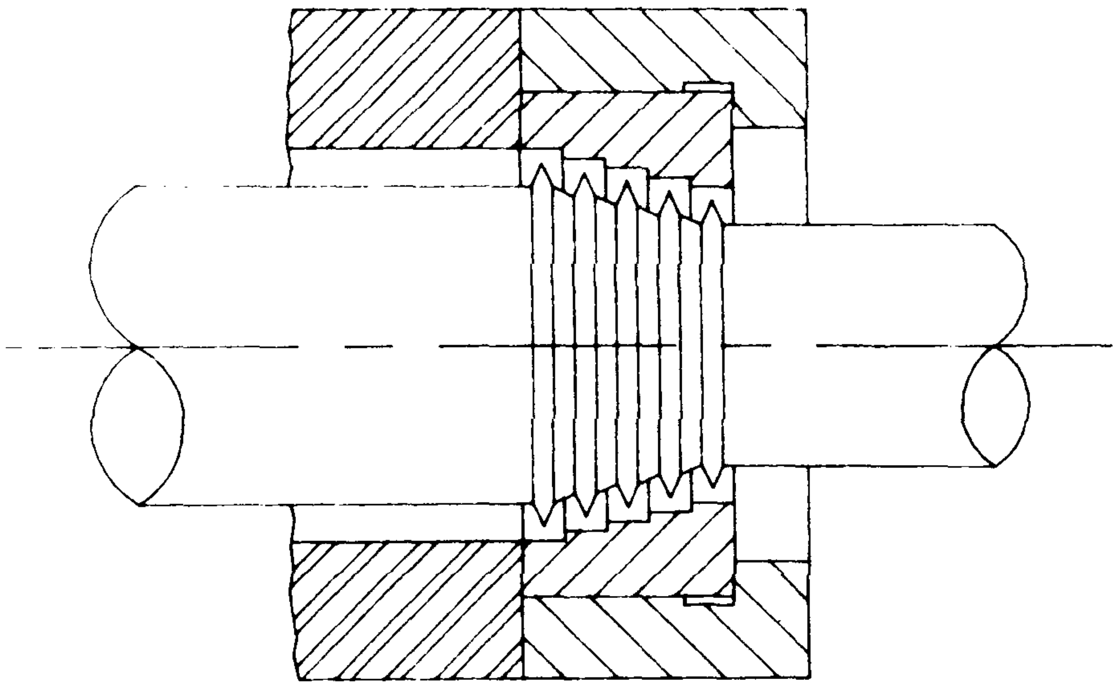
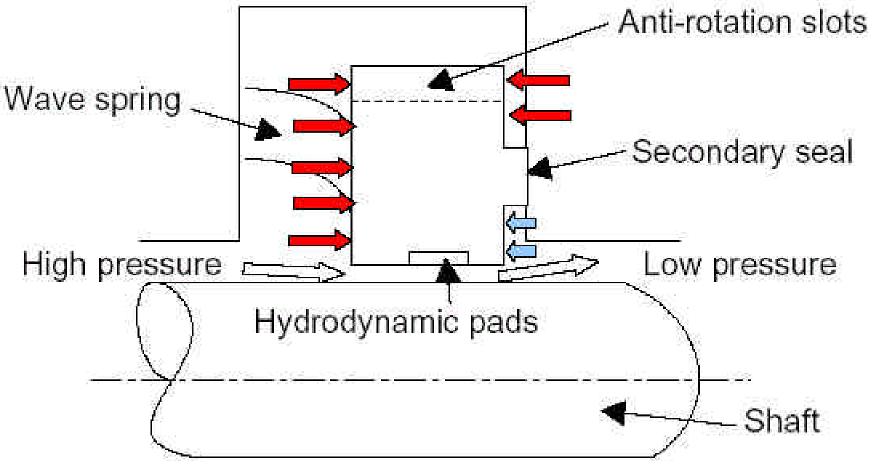


Figure 3 : Labyrinth seal



Dry gas ring seal configuration

Figure 4 : Ring seal

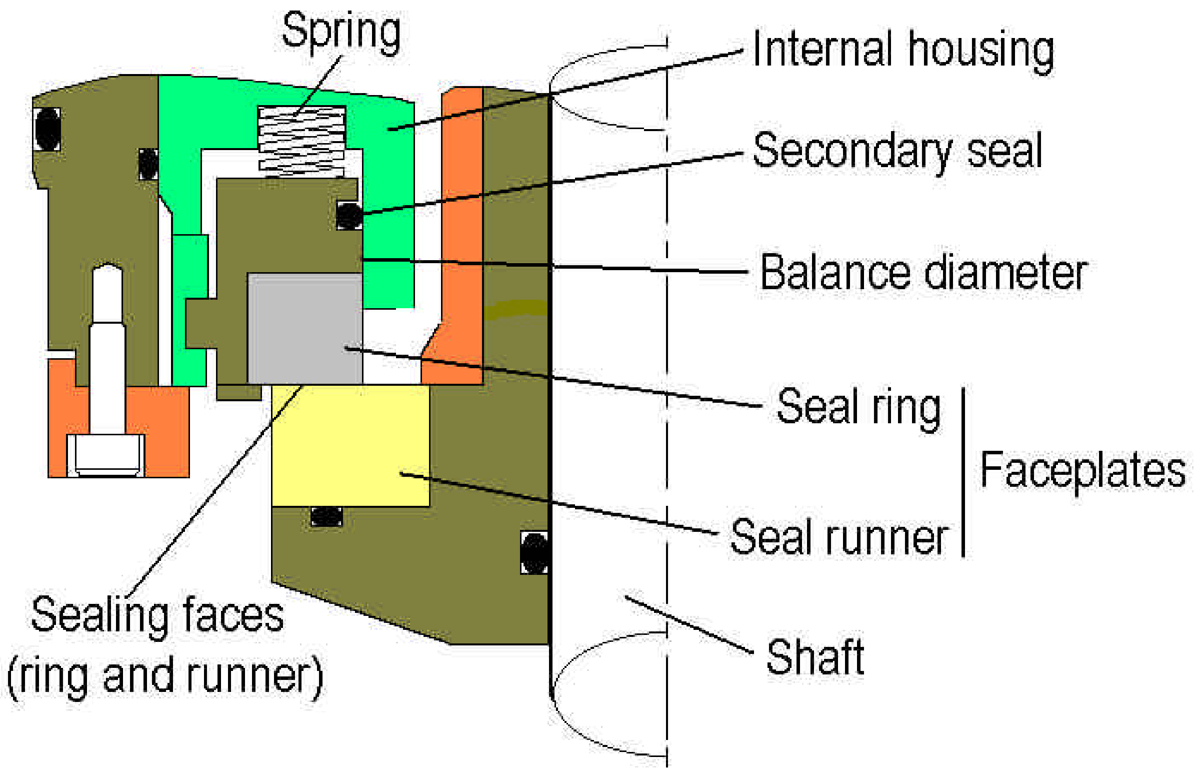


Figure 5 : Face seal

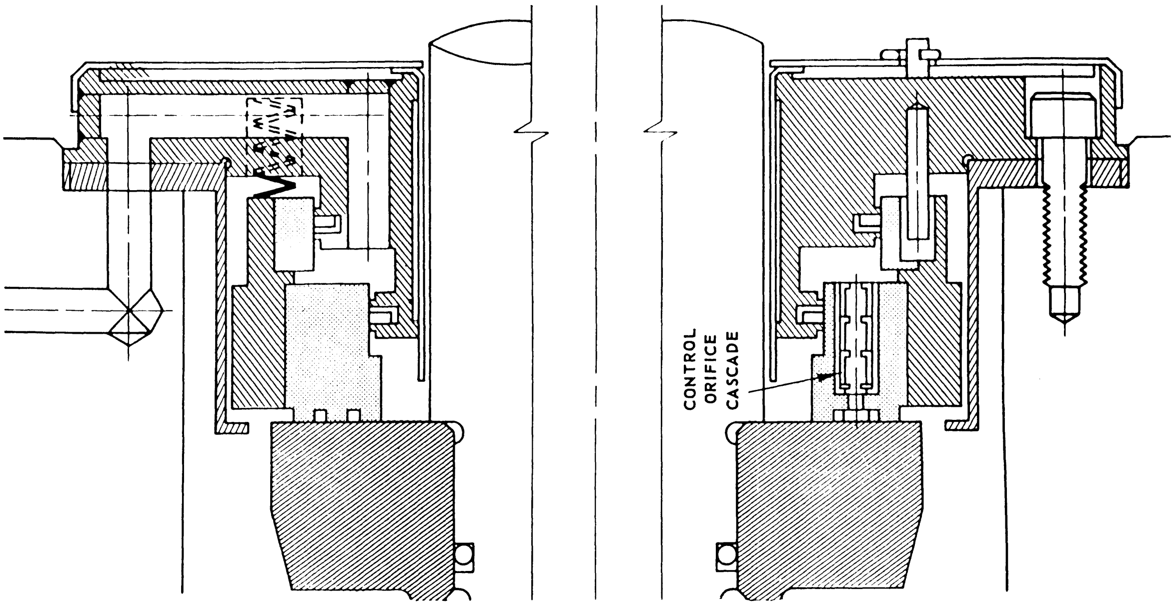


Figure 6 : Orifice hydrostatic seal

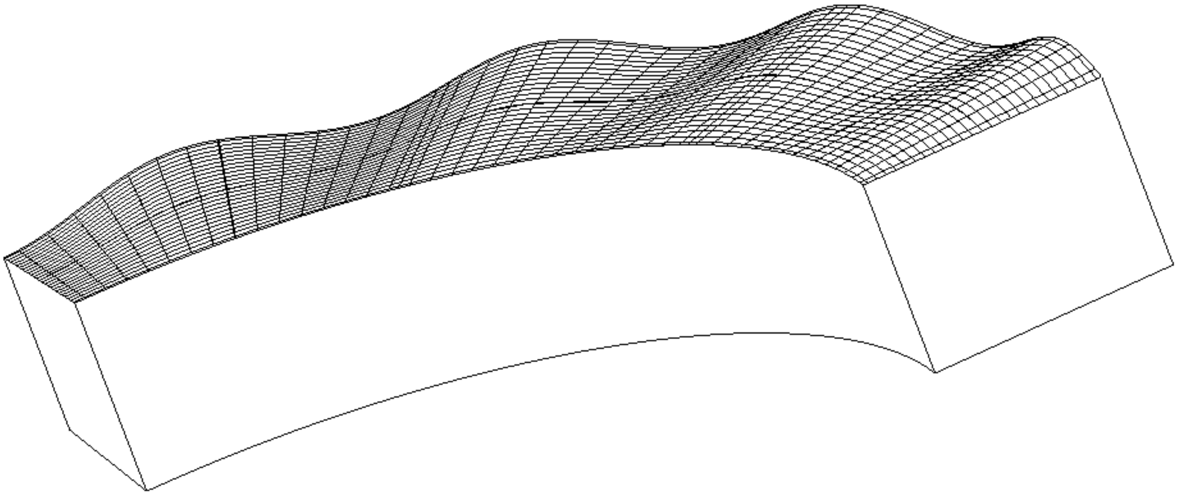


Figure 7 : Wavy sealing face

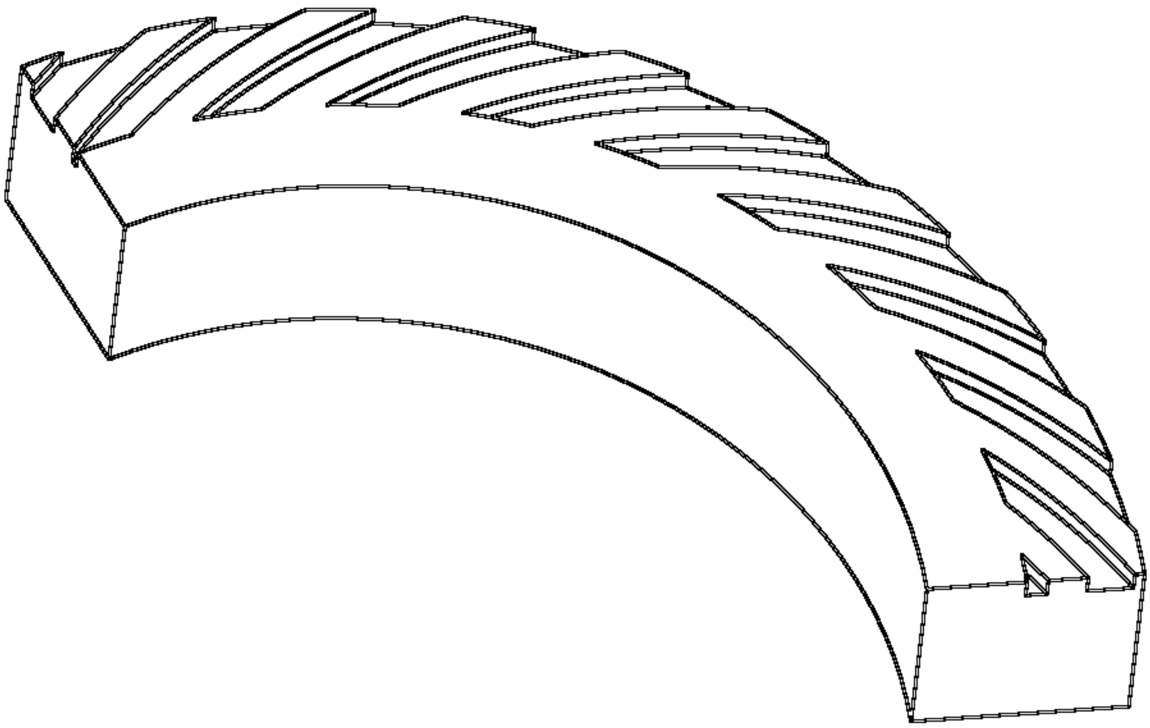


Figure 8 : Sealing face with spiral grooves

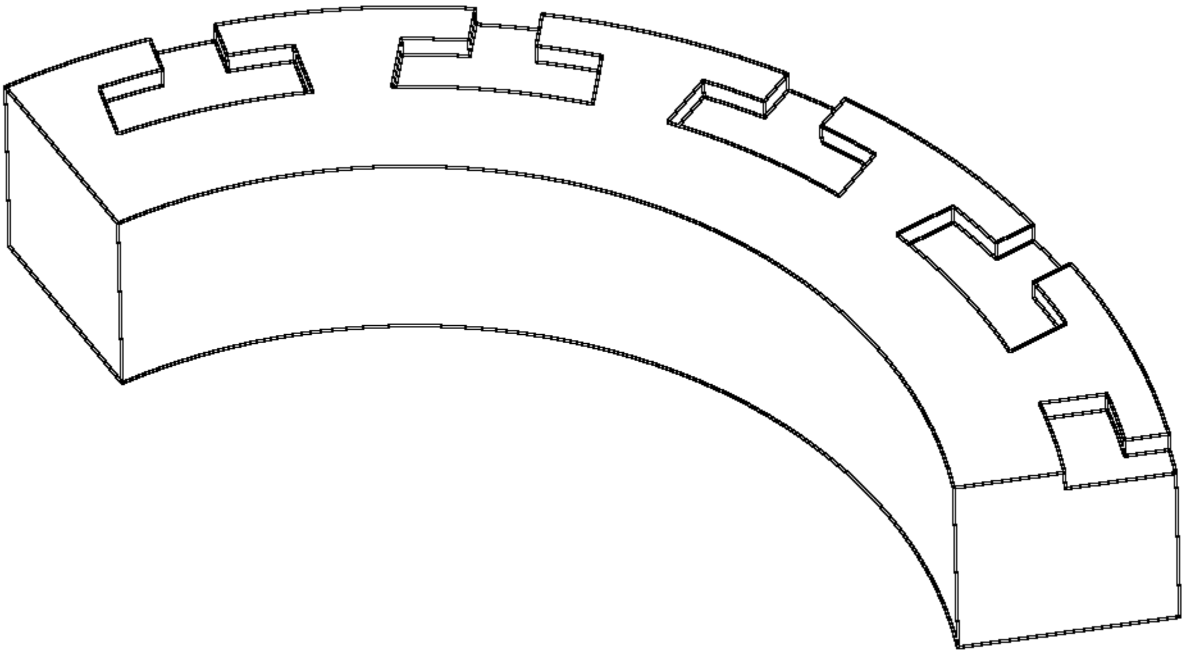


Figure 9 : Sealing face with symetric grooves (possible bi-directional rotation)

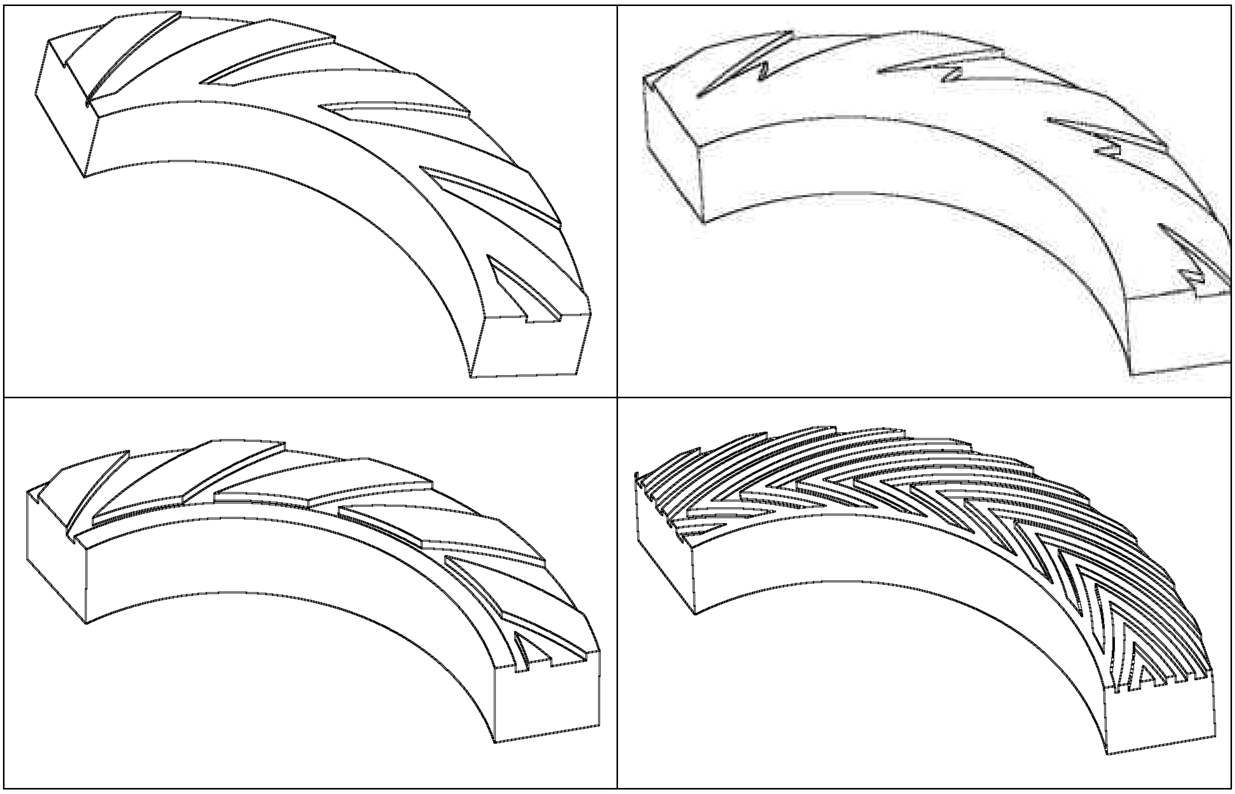


Figure 10 : Other grooved sealing face examples

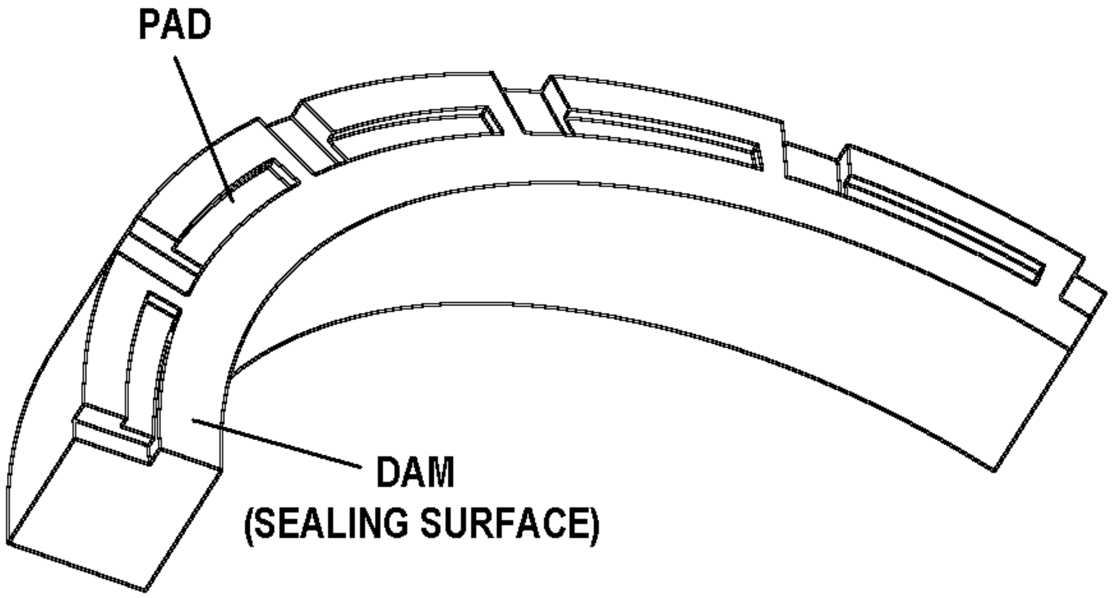
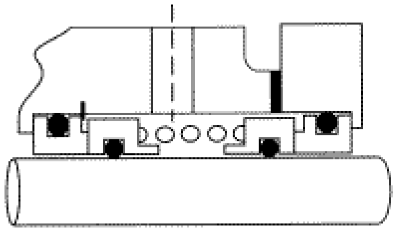
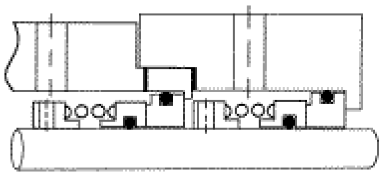


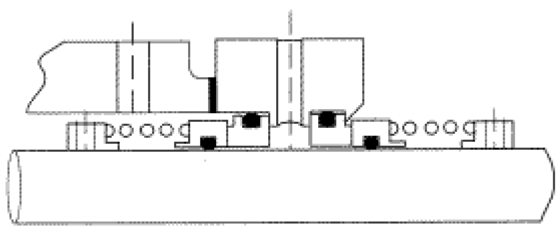
Figure 11 : Sealing face with Raleigh step pads



Back to Back



Tandem



Face to face

Figure 12 : Dual seal arrangements (stationary seals)

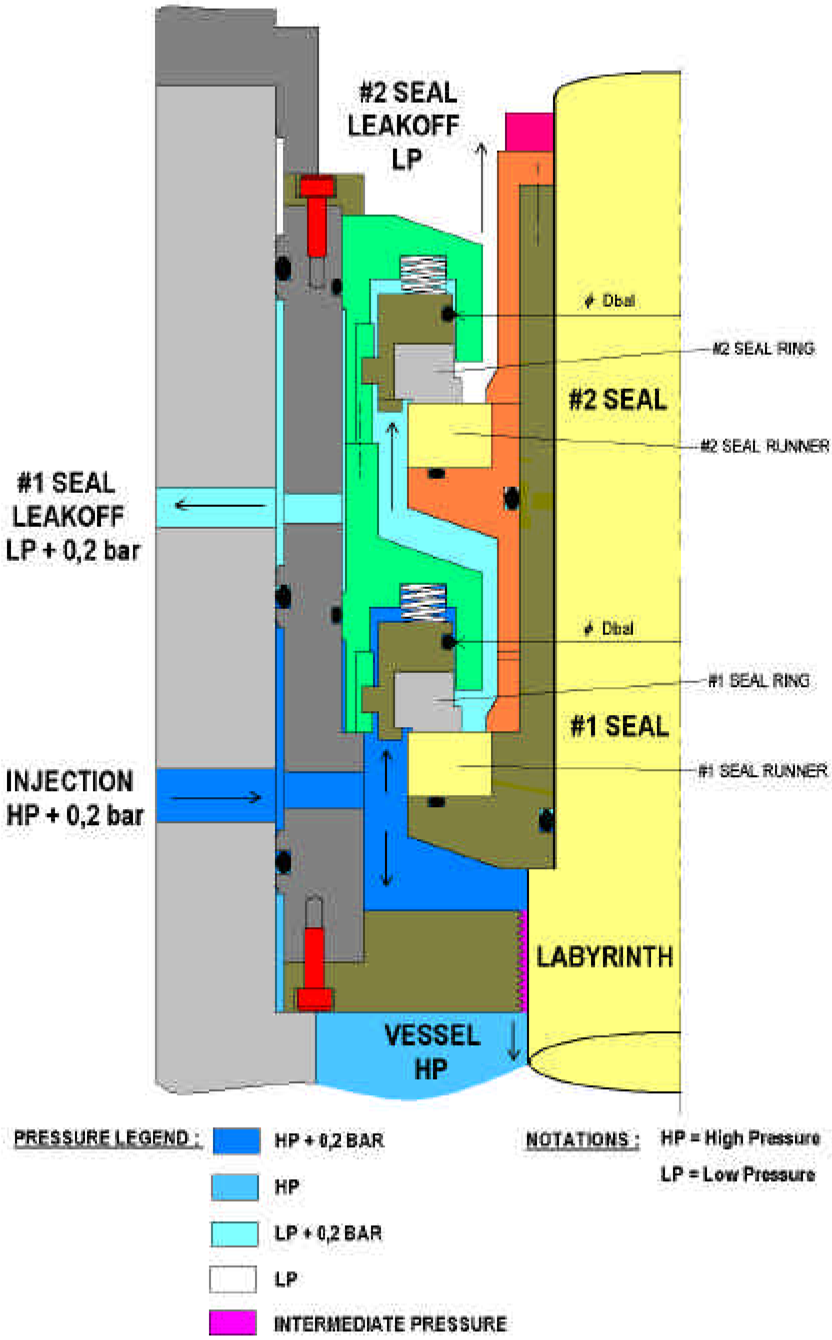


Figure 13 : Face seals assembly

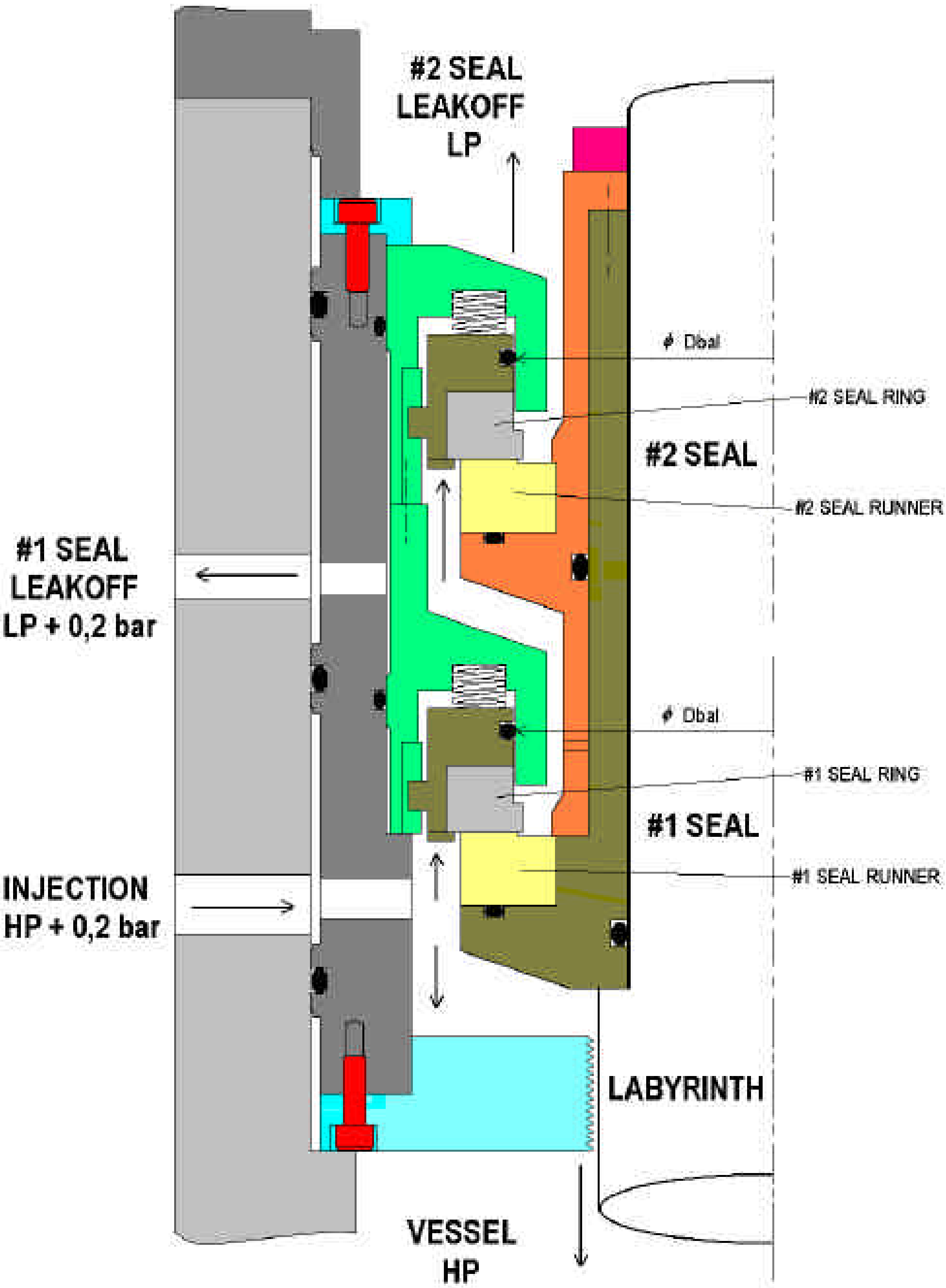


Figure 14 : Face seals assembly (for grooves on seal runner)

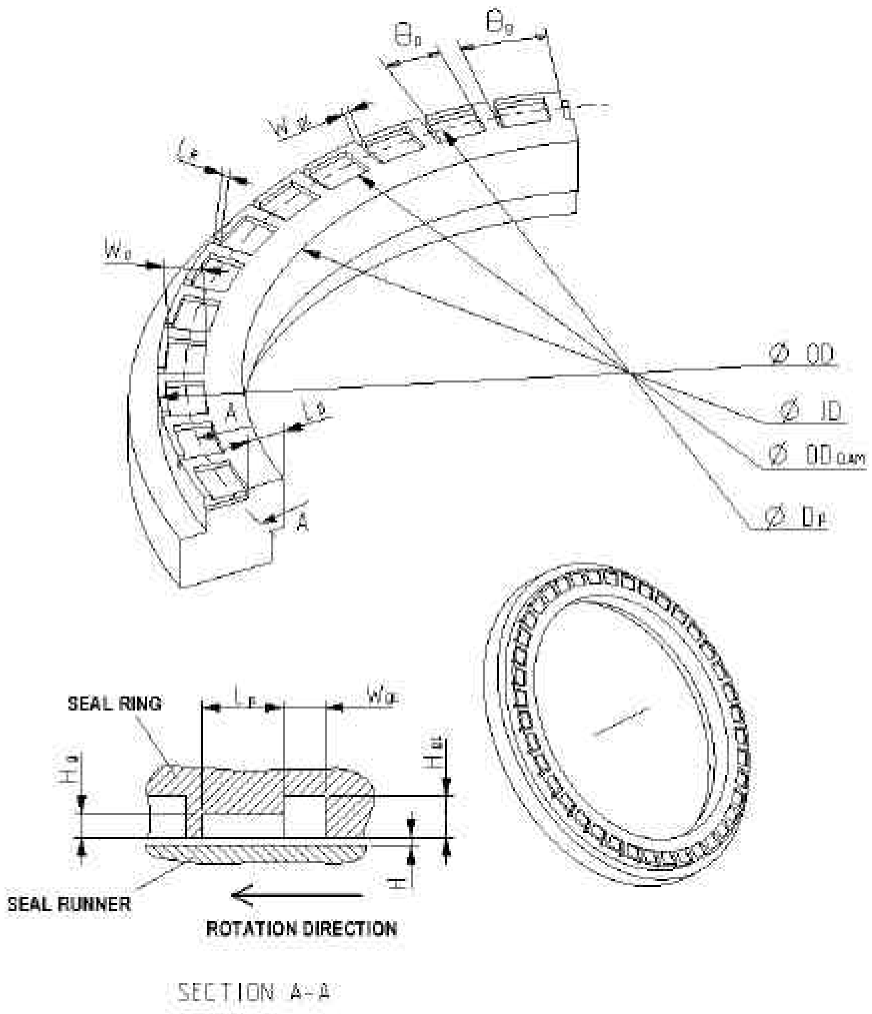


Figure 15 : Seal ring sealing face dimensions notations

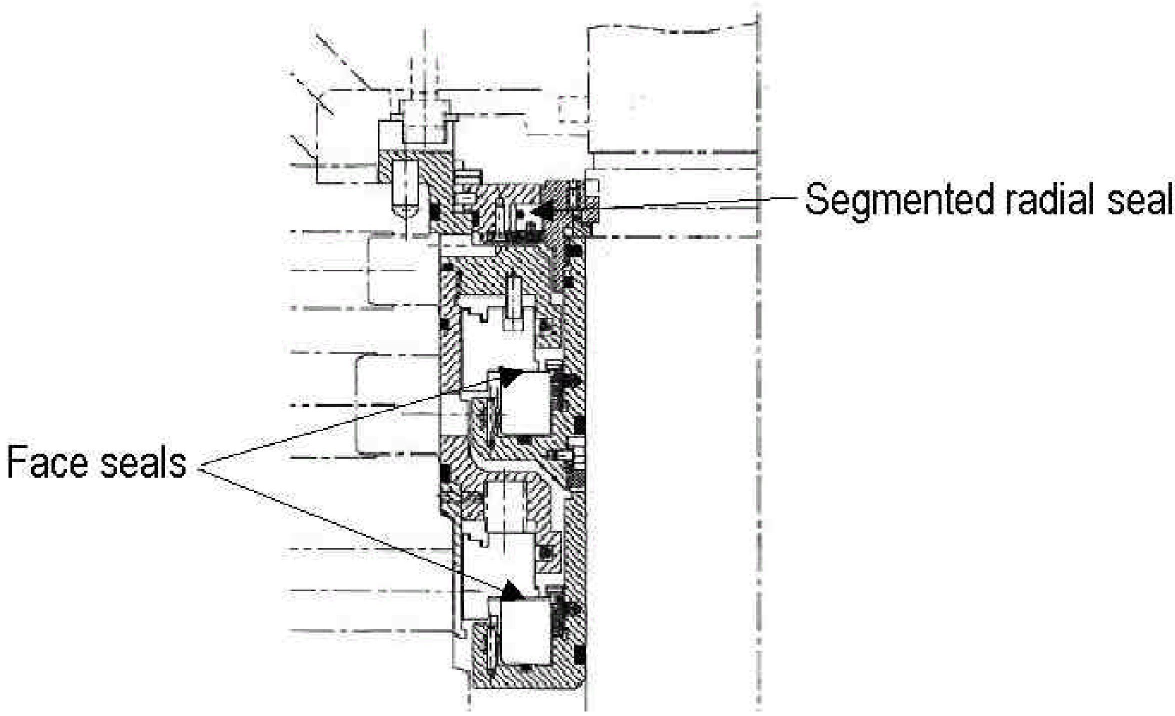
Note : Grooves depths are magnified on the figure to clarify the corresponding dimensions representation.

APPENDIX 1

FRAMATOME / JEUMONT DRY GAS SEAL

Jeumont developed a dry gas seal for compressor able to operate at a pressure of 110 bars and a rotation speed of 16000 rpm with a shaft diameter of 102 mm.

Because of the high pressure, the system is composed of two dry hydrodynamic face seals in tandem associated to a segmented radial seal.



The two hydrodynamic face seals includes :

- a silicon carbide seal runner
- a graphite seal ring with Raleigh pads on the sealing face

Each seal ring is balanced and submitted to the following forces :

- the hydrodynamic and hydrostatic lifting force in the interface,
- the seating force due to pressure on the seal ring other faces,
- the seating force due to springs between seal ring and its housing.

The secondary sealings are performed by means of carbon piston rings.

All the components are assembled in a cartridge for an easier and shorter installation.

The seal has been successfully tested at 14000 rpm with 110 bars on the first seal and 80 bars on the second seal. The maximum temperature in the chamber was 120°C, and the measured leakoff flow rate was about 13 Nm^3/h.

APPENDIX 2

FRAMATOME LE CREUSOT DRY GAS SEAL

Framatome tested a dry gas seal assembly for air compressor in its plant of Le Creusot in France.

The assembly was composed of a two hydrodynamic face seals of 100 mm diameter in tandem, a labyrinth between the compressor and the first seal and an injection of clean gas between the labyrinth and the first seal.

Each hydrodynamic seal was composed of a seal runner (rotating) with spiral grooves on sealing face and a carbon seal ring.

As in explained in Appendix 1 example, each seal ring is balanced and submitted to the following forces :

- the hydrodynamic and hydrostatic lifting force in the interface,
- the seating force due to pressure on the seal ring other faces,
- the seating force due to springs between seal ring and its housing.

This assembly was tested with the compressor working on a loop allowing to by-pass one seal and then to test a single arrangement or the tandem arrangement. Either nitrogen or carbon dioxide were used.

The tests were successful and did not require any particular precaution for compressor starting. The measured leakoff flow rates at 14000 rpm were :

- for a single arrangement : 5 Nm^3/h at 40 bars,
- for a tandem arrangement 10 Nm^3/h at 100 bars.