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Helium rotating seal
- Past experience in German HTR R&D Programme -

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1. Executive Summary

In 1974, a special rotating seal test facility was built by Brown, Boveri & CIE AG (BBC) in Kaefertal, Germany, within the frame of the "High-temperature reactor with Helium Turbine of big power (HHT)" Project. Two principle reactor designs as Integrated (INT) or non-Integrated (NINT) configurations were under discussion (see figures in Appendix). The NINT version resembles actual modular designs with separate vessels for the reactor and the power conversion system. All concepts had the electricity generator outside the primary system. Thus, the feasibility of a rotating seal was equally essential for this design option.

This test facility was operated to test the sealing of the rotor of the HHV plant, first, and then the sealing and components including the controlling of the total sealing systems for a nuclear power plant with direct cycle helium turbine.

The sealing system has to avoid ingress of air into the helium circuit and reduce the loss of helium from the primary side to acceptable limits.

As operational system, it was foreseen to use a swimming or floating ring with liquid barrier. For standstill a radially acting sealing element was used which could also accommodate an axial rotor shift due to assumed damages of the rotor bearings.

In the 1970th, no experience was available for rotor sealings of the requested size and for the high requirements with regard to availability, lifetime and low leakages.

Therefore, the test was performed in a 1:1 scale of the HHV plant rotor dimensions which also covered the conditions for the HHT prototype reactor. The rotor sealing had a diameter of 560 mm with three integral floating-ring sealings and three interlocking oil circuits. The rotational speed was 3000 rpm and the driving motor had a power of 450 kW.

The investigated media were nitrogen and helium till 50 bar. Pre-testing was made by nitrogen filling. Since Mid of 1974, the test facility was operated with helium.

Measurements include the throughputs and temperatures of the interlocking oil systems as well as the temperatures of the sealing devices. Results were used to optimise the tolerances of the sealings and to control the temperatures within the system.

The influence of rotor vibrations and of malfunctions in the oil systems were also investigated.

Due to the observed behaviour, many modifications of the original design were necessary. Thus, the testing took additional time and ended Mid 1975.

It was shown that the rotating seal was feasible under the design conditions of the HHV and HHT projects. Integral sealing rings would cause some problems in a nuclear plant for mounting and repair. Therefore, two or multi segment rings were recommended for future tests.

Figure 1 shows an overview on the complete test facility including the necessary peripherical supply systems. It also shows the complexity of the system which has to be taken into account when applying it in nuclear plants.

This kind of rotating seal was also tested full scale under operational conditions within the HHV plant. After overcoming initial problems leading to a significant oil ingress into the HHV gas circuit, it worked quite reliable.

It is recommended to modify the rotating seal in a way that oil ingress accidents can be excluded by design (e.g. combination with dry seals).

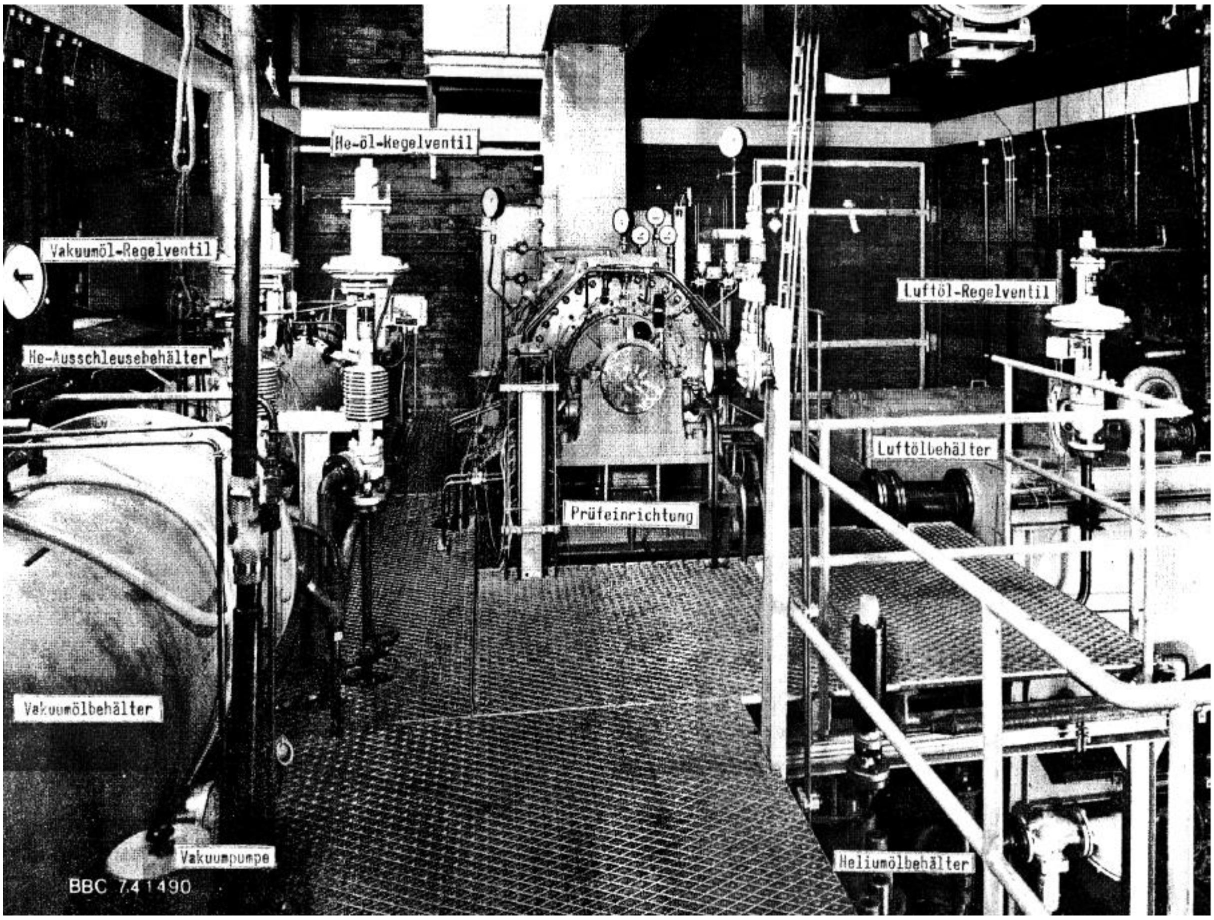

Figure 1: Test Facility for Rotating Seal at BBC (1975)

Legend:

Pruefeinrichtung	Test Facility
Vakuumoelbehaelter	Vacuum Oil Container
He-Ausschleusbehaelter	Helium Extraction Container
Vakuumoel-Regelventil	Vacuum Oil Control Valve
He-Oel-Regelventil	Helium Oil Control Valve
Luftoel-Regelventil	Air Oil Control Valve
Luftoelbehaelter	Air Oil Container
Heliumoelbehaelter	Helium Oil Container

2. Function of the Sealing System

The function of a swimming or floating ring sealing is based on the sealing effect of a fluid acting as a barrier within the gap between the rotating shaft and the swimming ring being movable in radial direction. The pressure of the barrier fluid (e.g. turbine oil) has to be higher than the medium (P_{system}) which has to be sealed. For the HHT project a rotating seal with three circuits has been chosen as operational system as illustrated in figure 2.

The three rings (Helium Oil Ring, Vacuum Oil Ring and Air Oil Ring) being situated one after the other between the primary gas side and the environment (air side) are fed by three separate oil circuits.

The two outer rings are provided with 0,8 bar higher pressure as compares to the system pressure and the one in the middle by 1 bar more from the vacuum oil circuit. This arrangement prevents that (contaminated) helium is transported to the air side sealing ring. On the other hand, it also hinders air being included in the air oil circuit enters the primary circuit causing e.g. oxidation of graphite.

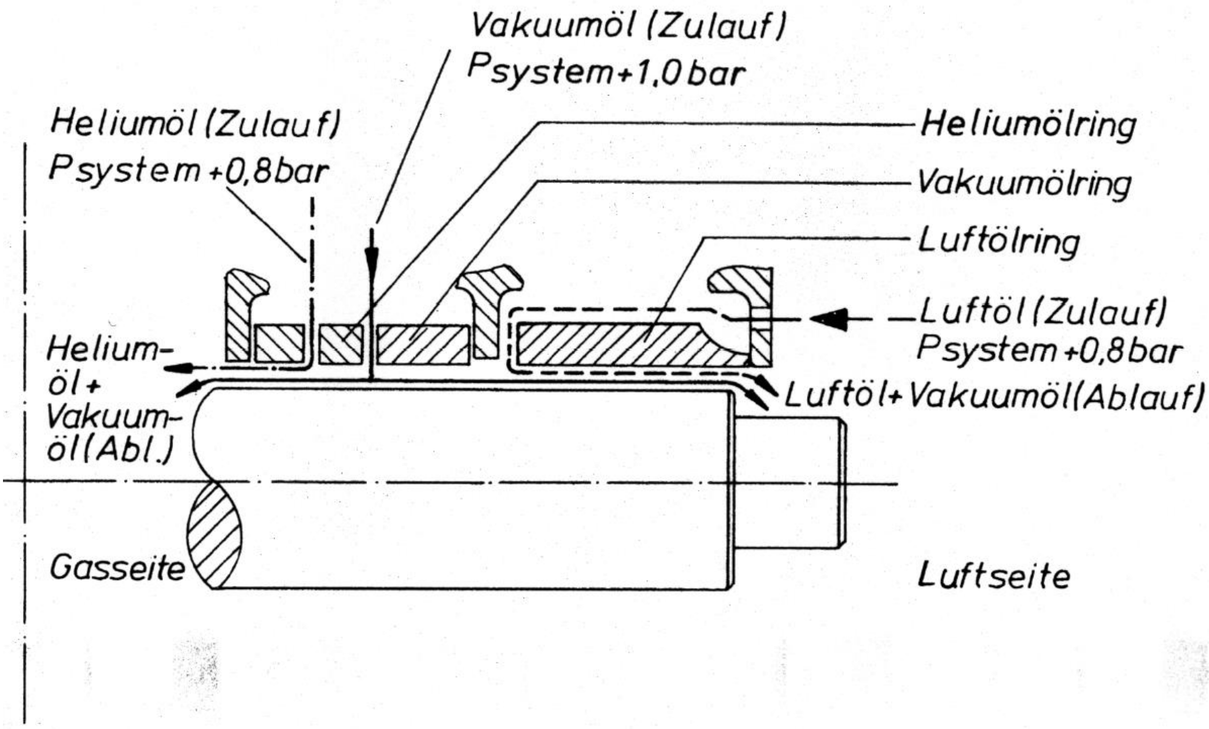

Figure 2: Functional Principle of Floating Ring Sealing with three Barrier Oil Circuits

The next figure illustrates the function of the three circuit barrier oil system:

The two floating rings on the helium gas side (01) are fed by a low pressure pump (05) from the helium oil container (04) which is under primary system pressure. In addition, there is a cooler (06) and a fine filter (07) in this circuit.

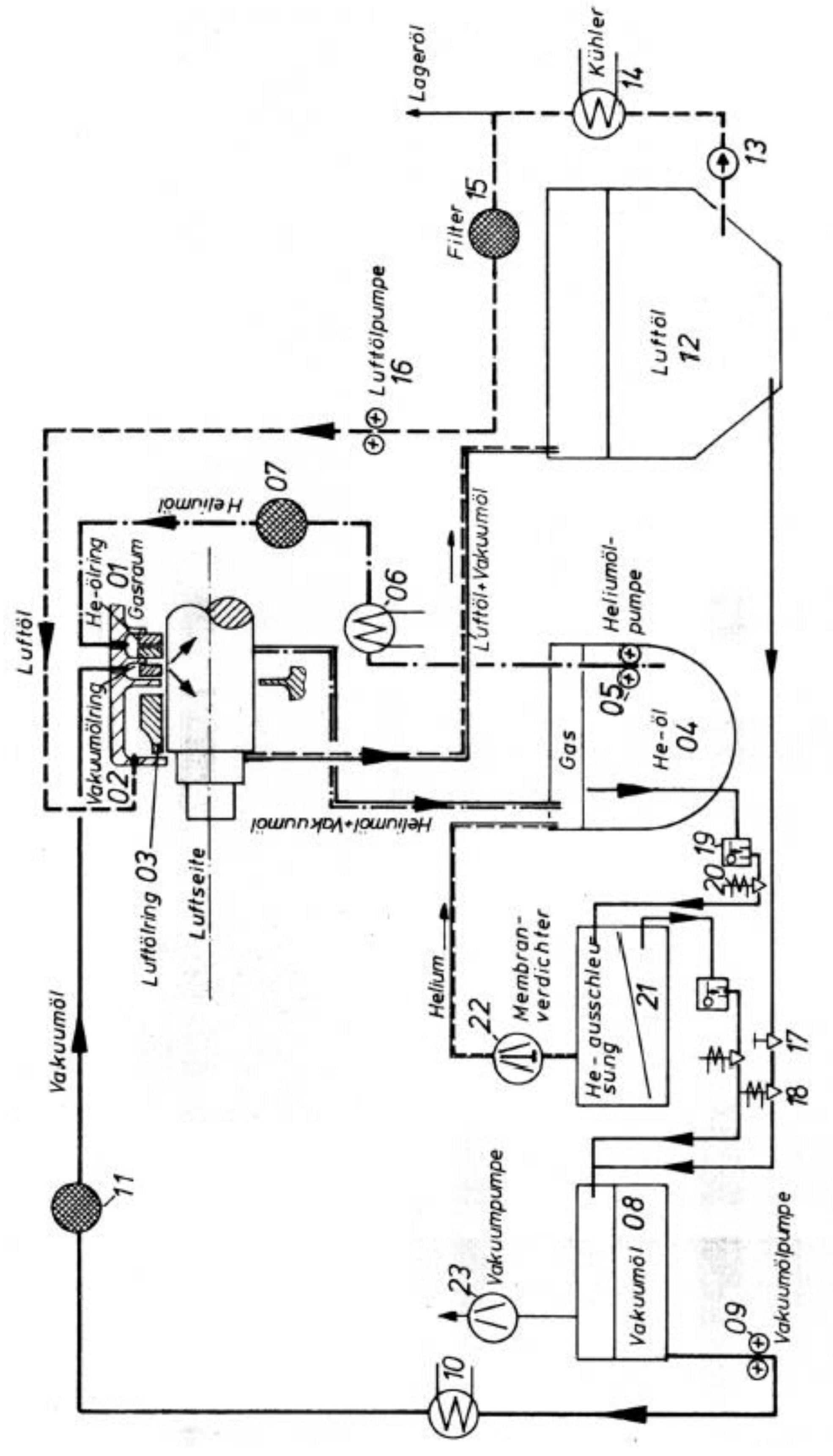

Figure 3: Oil Supply System

The two rings in the middle (Vacuum Rings) are provided by the high pressure pump (09) via a filter (11), the cooler (10) and from the vacuum oil container (08). The oil in this container is kept under vacuum to extract impurities. In an analogous way, the air side floating rings (03) are fed from the air oil container (12) via the auxiliary pump (13), the cooler (14), filter (15) and high pressure pump (16).

Helium oil and vacuum oil are fed back to the helium oil container. A part of this oil is then transferred to a container where the helium is removed and pumped back to the helium oil container by a membrane pump (22). This process is controlled by a pneumatic control valve (19) and a magnet valve (20).

In a similar way the fraction of the air and vacuum oil flows back to the air oil container (12). A part of this oil is fed back to the vacuum oil container controlled by two valves (17 and 18).

This description shows that this kind of rotating seal follows simple principles but needs a rather complex support systems. It is also obvious that the tuning of the pressure differences and gaps between the rings needs special attention. This was done in the special test facility as illustrated in more detail in the next chapter.

3. Description of the Test Facility

The test section consists of a rotor being suspended in two oil lubricated floating bearings. The rotor has a length of 1841 mm and a diameter of 560 mm which is quite representative even for the GT-MHR design (>450 mm) or PBMR (~ 350 mm). Symmetrically, there are two rotating seals in the form of three undivided floating rings. The rotor is driven by a 450 kW direct current shunt motor with variable speed.

The rotor and sealing systems are surrounded by an integral cylindrical housing. In the interspace between the sealings there is a gas volume for nitrogen or helium with a pressure up to 52 bar. The symmetrical arrangement avoids axial thrust.

The floating rings on the gas side (01) as well as the middle ring (02) are made of bronze and are screwed together. Oil supply is made for 01 by radial boreholes and via a gap between the rings for 02.

The air ring is manufactured from steel and filled with white allow (Babbitt).

The two inner rings only bear little pressure differenced but the outer one keeps the full pressure between primary system and atmosphere plus the operational pressure of the oil circuits. Therefore the outer ring is much broader as compared to the others.

In addition to the rotating seal there is a sealing for standstill as illustrated in figure 5. It mainly contains of symmetrical sealing elements with a piston ring for axial closure of the gap. Pressurised oil or gas is injected between the pistons to activate the system. A further radial gum sealing ring can be expanded to close the boreholes for pressure adaptation. Opening and closing of the standstill sealing is made by hydraulic pumps. Springs keep it open under rotation of the shaft.

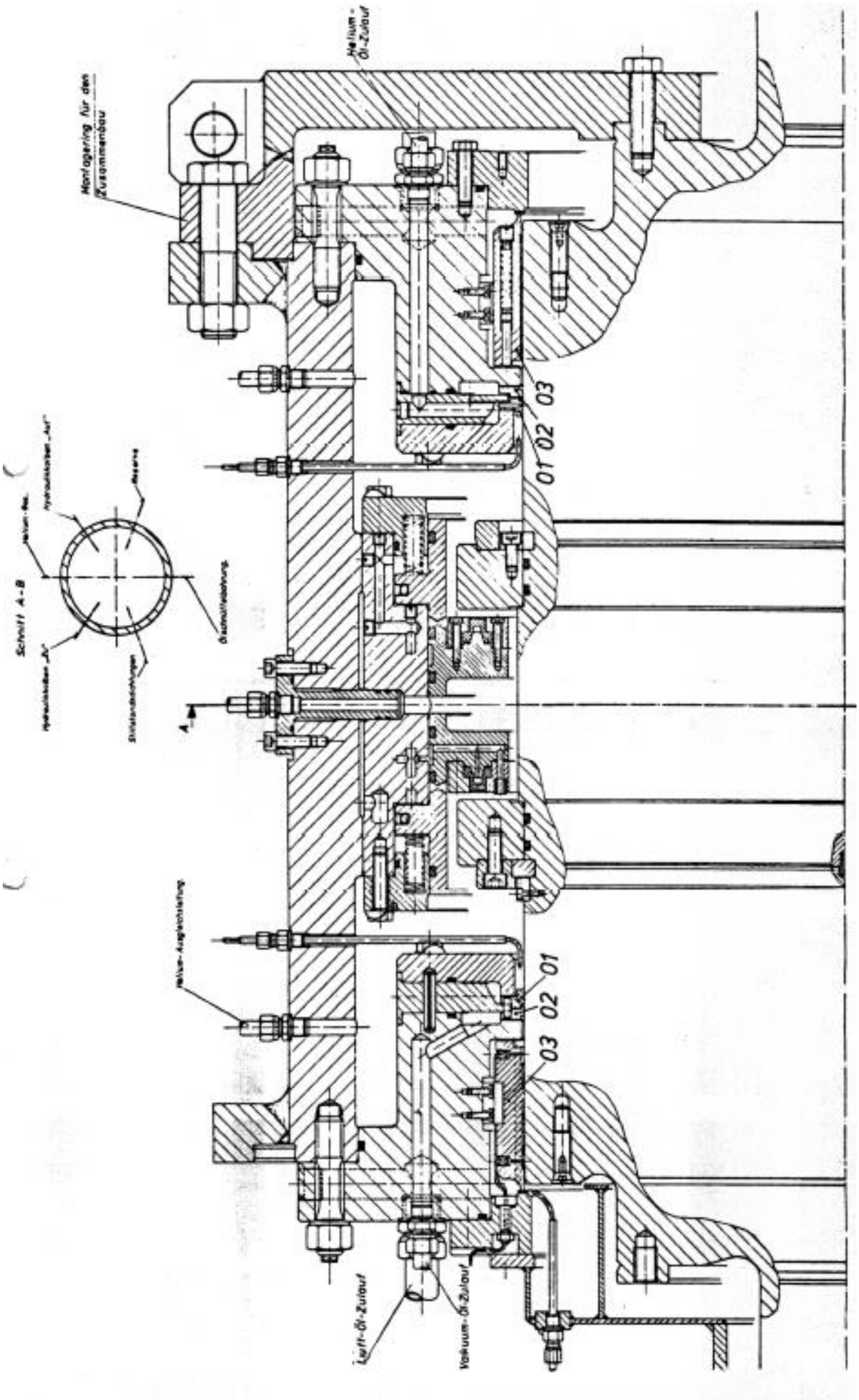

Figure 4: Cross-Section of Test Arrangement

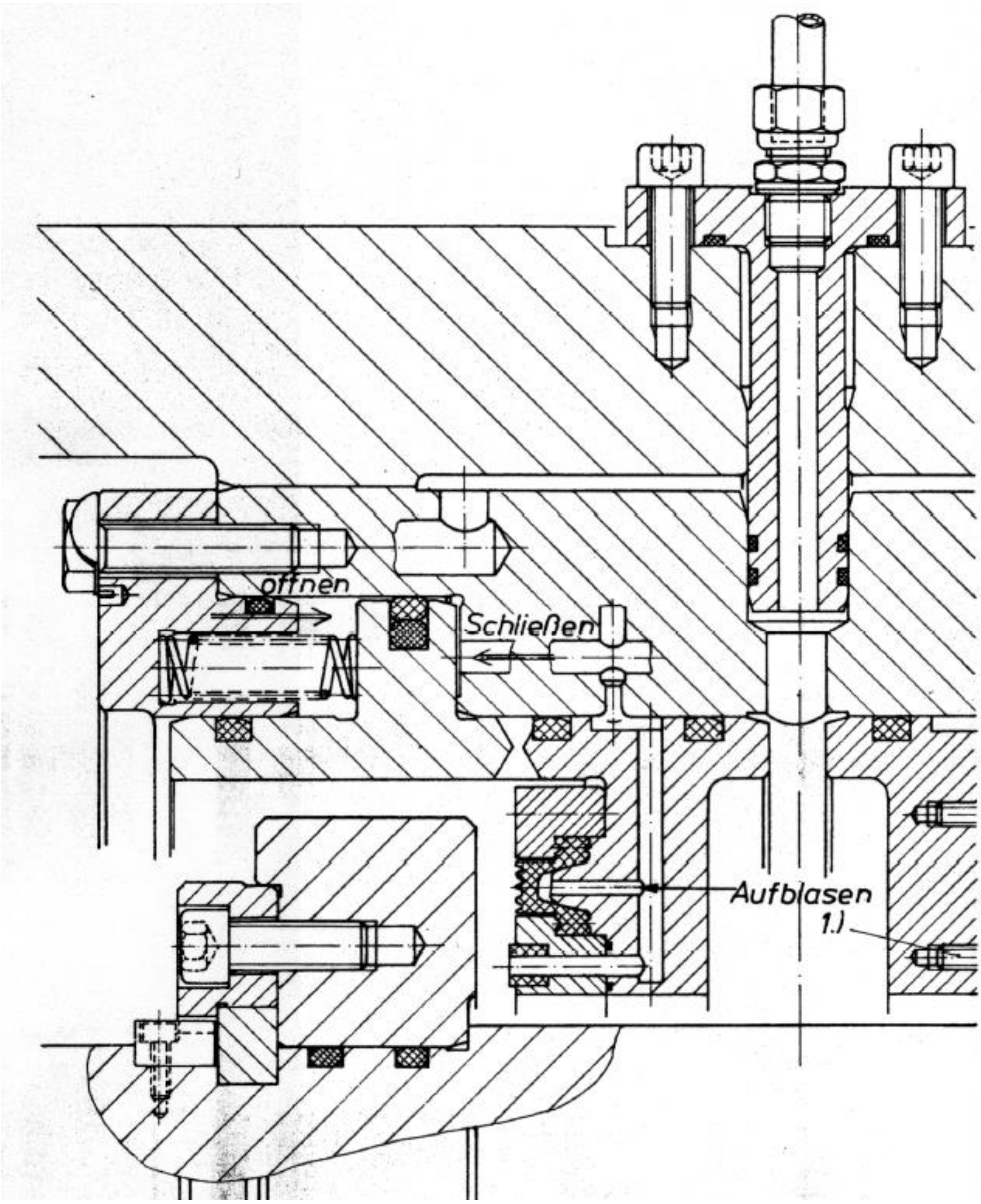

Figure 5: Sealing for Standstill of Rotor

4. Control and Safety Installations

The control of the differential pressures is done by a pneumatic control system. Pressure differences are measured, transformed, amplified and used to activate membrane control valves which feed excessive oil to the different oil containers.

Temperatures are controlled by variation of throughputs through the coolers within the coolant water circuits. In addition, there are heaters in the air oil container and on the helium oil pressure tube being controlled by thermostats.

All oil systems are sheltered by overpressure valves.

Control of oil level is made by floating switches. System pressure, oil pressure in the bearings and differential pressures across the filters are monitored to initiate shut down of the facility in case of leaving allowable values.

Temperatures are measured in the floating rings, in the bearings and in the different oil circuits. The correct function of all pumps is surveyed to allow for start-up or activate shut-down in case of malfunction.

5. Measuring Equipment

The focus of the measurements during the experiments was

- Throughput of barrier oil through the gaps of the floating rings
- Absolute and differential oil pressures in the sealing system
- Oil temperatures in the different parts of the oil circuits
- Babbitt metal temperatures
- Gas temperatures
- Power of the driving motor

Throughputs were measured by turbine inducers whereas absolute and differential pressures were detected by tube spring or dose manometers. Temperatures were measured by pointer and glas thermometers as well as by thermo elements. Especially the floating rings were surveyed around their periphery each 120 degrees.

From today's point of view, all measuring and recording equipment could be subject of a technical museum but nevertheless allowed an exact measurement of the principal function of the rotating seal and to provide indications for their optimisation.

6. Test Results

6.1 Preliminary tests with nitrogen

From cost reasons, the pre-testing was planned to be done with nitrogen. This allowed for checking the principal function of the overall system.

However, tests with increasing pressures had to be interrupted due to unacceptable high temperatures of the air oil. Therefore, the gap between the air oil ring and the rotor had to be increased from 0,4 to 0,5 mm to enhance the oil throughput and to lower the temperatures. Nevertheless, a system pressure of 5 bar at 3000 rpm could not be surmounted due to the high oil temperatures.

The conclusions from these observations are also relevant for future tests of rotating seals as nitrogen has a seven times higher density and generates a seven times higher friction heat as compared to helium. This fact leads to rather high heat transfer to the rotor and reduces oil viscosity due to the higher temperatures.

In addition, excessive foam generation was observed in the helium extraction container. Therefore, back-feed of oil was hindered due to the gas bubbles and malfunction of floating switches. The reason was the high solubility of nitrogen in oil.

Under these conditions, it was decided to give up further experiments under nitrogen and to continue testing with helium. These difficulties should be respected when planning for new tests of rotating seals which will only be representative when done under helium atmosphere.

6.2 Tests under helium

After changing the atmosphere to helium, the facility could be operated stationary over longer time periods in a way that the main parameters could be determined in dependence of the system pressure at 3000 rpm. The differential pressure between air oil and system oil was about 0,8 bar with a gap under cold conditions of 0,5 mm.

However, the measured throughputs are much lower than the calculated values and deviate significantly with increasing system pressure. This is due to the fact that even under helium, friction work between rotor and stationary gas is increasing in a way that the rotor temperature and its diameter are raising. The floating rings are cooled by the fresh oil and do not expand that much. Consequently, the gaps and the associated throughputs are getting smaller.

The Babbitt metal temperatures reached critical temperatures on the oil exit side. Thus, the gap of the air oil ring had to be increased a second time by another 0,1 mm. After this change, the Babbitt metal temperatures came down to tolerable values.

Under low rotational speed (<50 rpm), some eccentricity of the rings could be observed. The sickle-shaped gaps lead to some increase in oil throughputs.

With low system pressures and cold rotor the larger thermal expansion of the bronze rings could be observed leading to larger gaps. At higher pressures, the effect was reduced by the increase of rotor temperature.

The rotor speed had some influence on the oil throughput of the helium oil ring and the air oil ring which are broader than the vacuum oil ring. This leads to higher friction and temperature increase with higher speed for these rings.

The experiments have shown that there is a rather complex interaction of the different components of the rotating seal and that it needs very careful design and testing under representative conditions before applying to a nuclear system.

6.3 Accident simulation

It was investigated whether the rotating seal could be kept functional if the helium oil circuit or the vacuum oil circuit (or even both) fail. The air oil circuit must not fail at all as it has to seal against the full pressure difference. The vacuum and helium oil circuit pumps are not sufficient to provide sealing under those conditions. In addition, the mechanical strength of the bronze rings does not allow them to take the complete pressure difference.

In the HHV plant, the air oil supply was secured by emergency electric current and containers on a higher elevation to take care for a safe run-down in case of disturbances in the sealing system.

Under operational conditions (50 bar / 3000 rpm), shut-down of the helium oil pump lead to an increase of vacuum oil throughput from 18 to 32 l/min. This means that the missing helium oil was replaced by additional vacuum oil. The differential pressure at the helium oil ring kept at 0,12 bar. The air oil throughput was not influenced at all and temperature increased only by 5°C.

Failure of the vacuum oil pump would have lead to problems in managing the oil levels in the oil containers. In addition, the vacuum ring would be not sufficiently provided by oil if the pressures of the helium oil and the air oil is nearly equal. Therefore, a test has been made only with a helium oil pressure being 0,2 bar below the air oil circuit pressure. Missing vacuum oil is now substituted by air oil. The air oil throughput increases by 20 l/min and kept a differential pressure of 0,7 bar across the vacuum oil ring. Temperatures kept constant. However, such conditions would not be tolerable in a nuclear reactor as dissolved air would be transported into the primary system.

Simultaneous failure of the helium and vacuum oil pump leads to rather low throughputs at the inner floating rings which do not allow to dissipate friction heat. Besides that, the differential air oil pressure is reduced to zero in about 5 sec till activation of the control system. For a short time, there is the danger of a system gas ejection.

Such an effect could be avoided by enhancing the level of the differential pressures by 0,2-0,4 bar. This measure would not have a strong influence on the oil throughputs but could improve the overall emergency response of the rotating seal.

6.4 Tests under rotor vibrations

For testing the rotating seal in case of rotor vibrations an eccentrical weight of 20 g was mounted to the rotor on both sides of the bearings on the same angle about 250 mm from the middle axis. This resulted in 3-4 micrometer amplitude at the foundation of the bearing. It is assumed the rotor amplitude was 7-10 times higher. A direct measurement was not possible.

No negative effect was found on the function of the rotating seal and the control of the sealing system. This was still the case when increasing eccentrical weights to 75 g resulting in amplitudes in the bearings to 14 micrometer instead of 3-4 micrometer.

7. Test of stand-still sealings

The functional principle of the stand-still sealings has been explained in chapter 3 and figure 5. For the test of this sealing, it had to be guaranteed that the pistons are simultaneously activated to avoid an axial thrust damaging the roller bearings of the motor. This is specific to the test facility because the forces of the sealing could be easily adsorbed in a large turbo generator.

Figure 6 shows the time response after actuation of the control valves. When closing the sealing, the oil pressure increases to about 2 bar to overcome the spring and friction forces. During the closure process of about 90 sec, pressure still slightly increases till reaching the end position.

When expanding the redundant gum sealings, pressure decreases a little bit for a short while down to 2 bar.

Opening is supported by the springs and starts at 1,5 bar and is completed after 45 sec.

After closure, the system has been charged by 39,6 bar with helium. All connections of the test vessel had been closed for this test to avoid parasitical leakages. After 44,5 hours, the pressure dropped down to 38,8 bar corresponding to a leak rate (L) of

$$L = 6{,}75 * 10^{-2} \text{ Torr l / sec}$$

The pressure drop in the hydraulic system during this time was about 0,3 bar. This indicates that larger oil leakages did not occur.

8. Long-duration test

Under a system pressure of 49 bar and a speed of 3000 rpm the facility has been operated about 50 hours without interruption.

For providing enough air oil, the air oil reserve pump has been operated from the very beginning to avoid instabilities when starting the pump during the test or to activate safety valves as observed during a similar test.

After 5 hours, the facility reached stationary conditions with regard to temperatures and oil throughputs. At 30 hours the power of the membrane compressor 22 slowly degraded. The reasonwas some ingress of oil in the first compression stage and some air in the hydraulic system. These problems can be avoided by improving the oil separation in this system.

During the test, the helium pressure decreased by about 0,5 bar per hour. As leakages could not be detected, the reason has been seen in a not complete degasing of the vacuum oil.

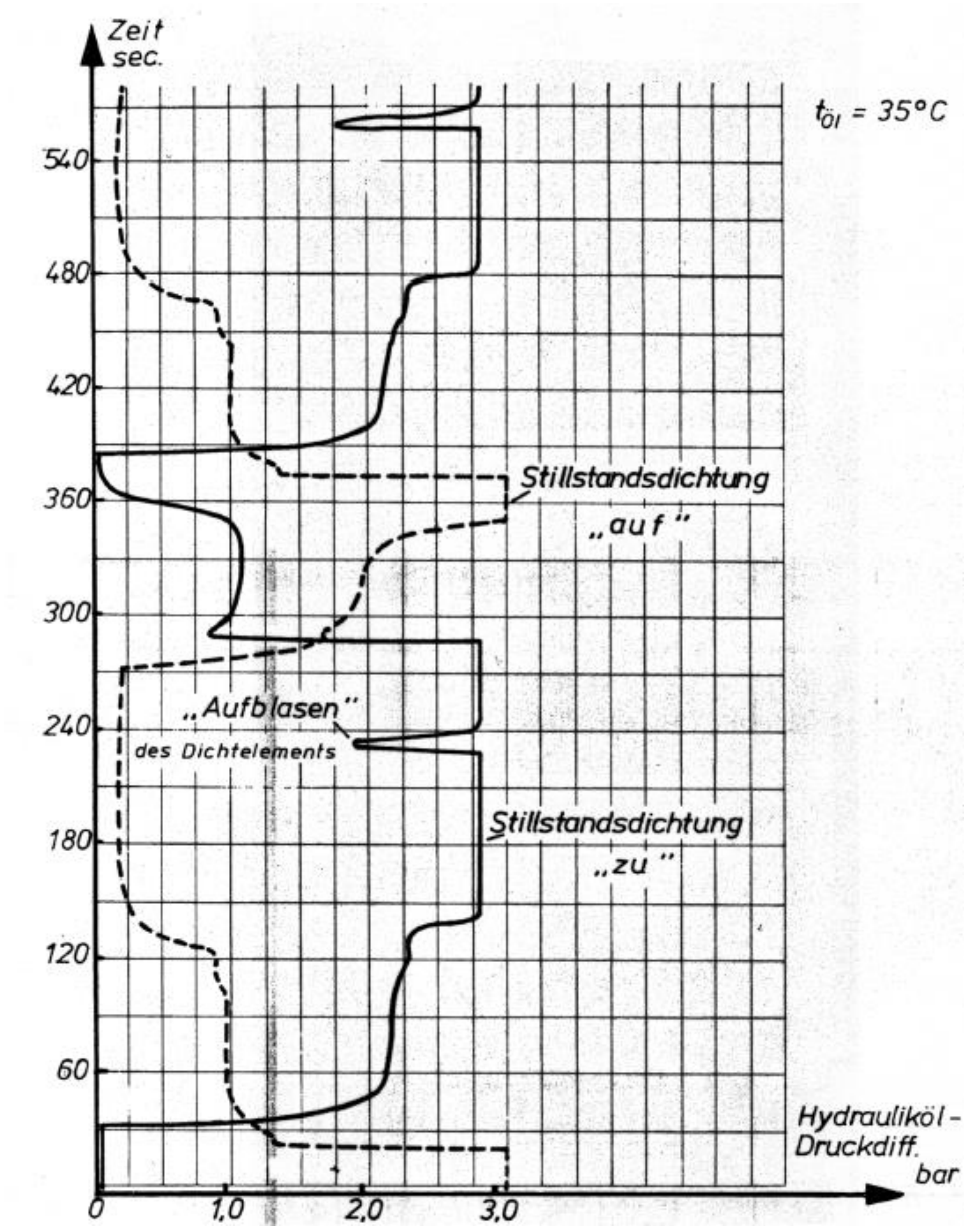

Figure 6: Actuation of Sealing for Standstill of Rotor

Legend:

Stillstandsdichtung	*Sealing for Standstill of Rotor*
Aufblasen	*Pressurisation of Sealing*

9. Conclusions

In summary, the tests have shown the feasibility of the rotating seal on the principle of floating rings and provided fundamental data for the optimisation of the system.

The problems which have been observed are not caused by the basic design but mainly by the rather complex auxiliary system providing the oil for the three separate circuits and the management of the oil volumes in the fluid containers. This can much more reliable be handled by today's electronic control systems.

Integral floating rings may be difficult with regard to mounting and repair. Thus, divided rings might be more preferable.

Caused by a malfunction of the rotating seal, an unintended large oil ingress occurred during the commissioning HHV plant contaminating the whole gas circuit. This event also shows that such rotating seals are rather sensible with regard to the complex auxiliary system providing the oil for the different circuits.

The test results also show that the interaction of the mechanical parts, their different thermal expansion and the overall thermal hydraulics of the rotating seal has to be carefully investigated.

Appendix

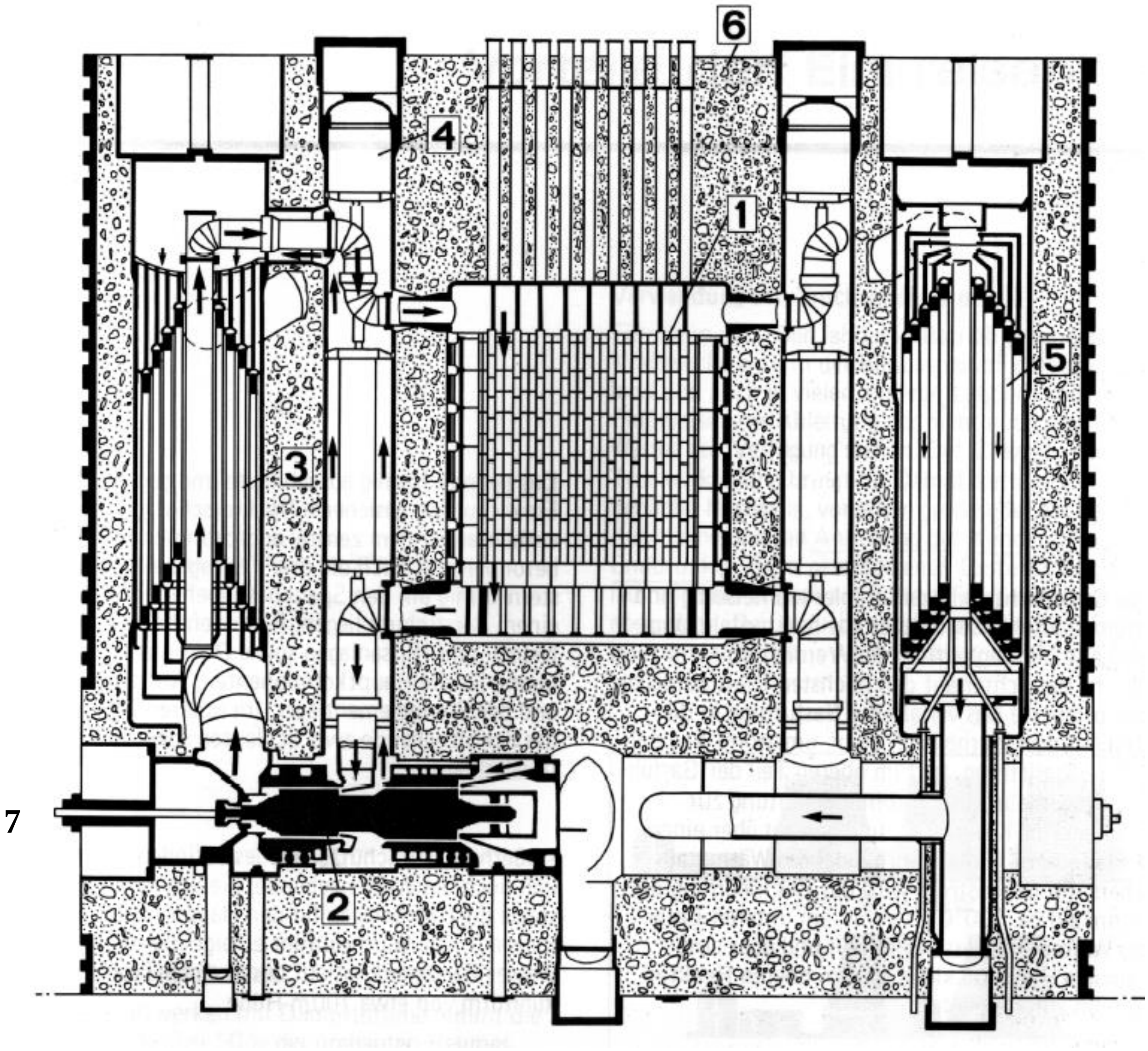

Figure A1: Cross Section of Integrated HHT Reference Plant Design (INT)

Legend:

1	*Reactor Core*
2	*Gas Turbine*
3	*Recuperator*
4	*High Pressure Caverns*
5	*Cooler*
6	*Prestressed Concrete Pressure Vessel*
7	*Shaft to Generator with Rotating Seal*

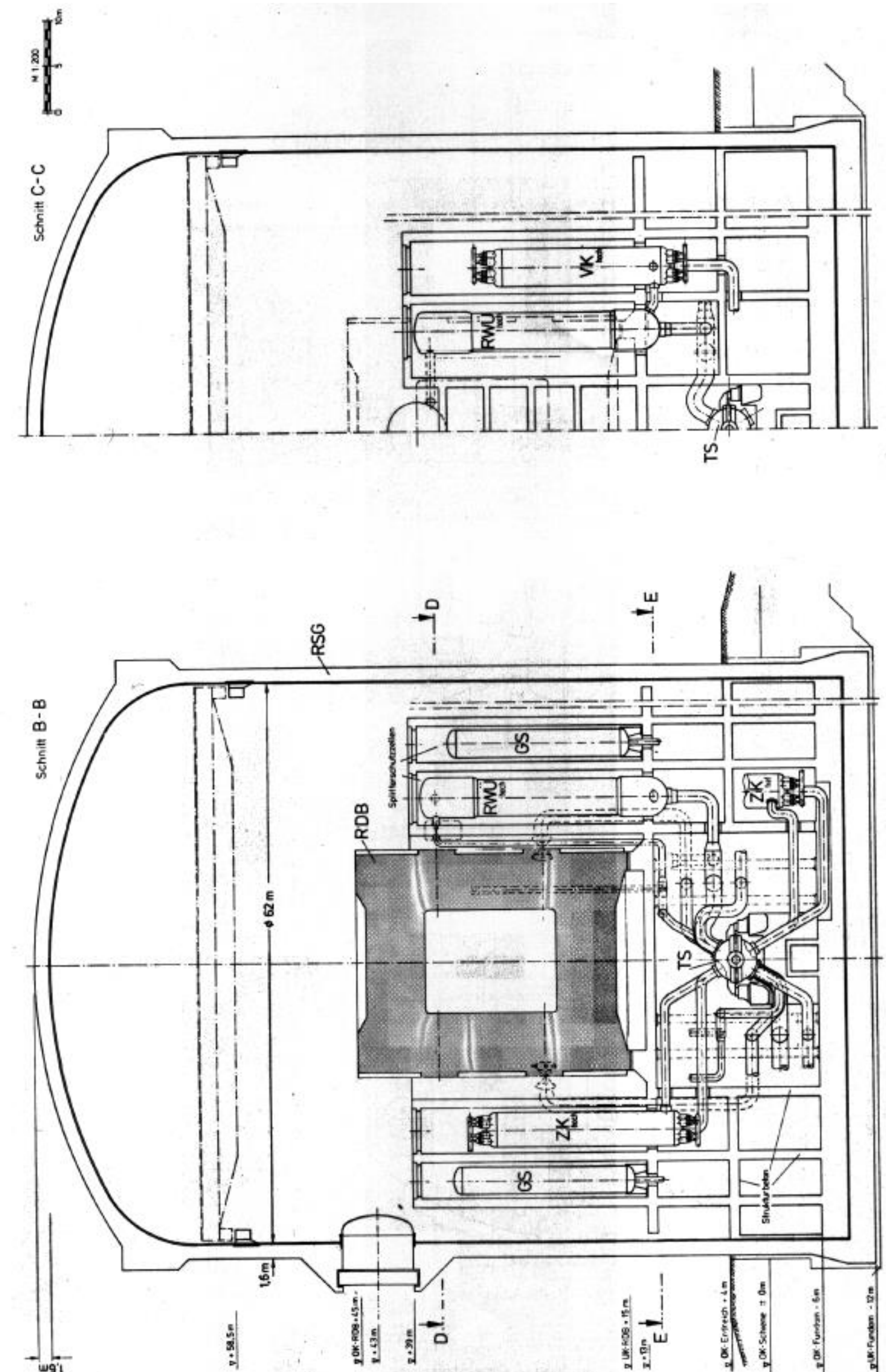

Figure A2: Turboset (TS) in Non-Integrated HHT Design (NINT)

References

The report is based on the following documents:

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