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WP3 Magnetic bearings and catcher bearings
Functional requirements

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Summary

This document contributes to the 5th PCRD HTR-E WP3. It provides the functional requirements for axial and radial bearings supporting the Turbo-Machine (TM) for 2 HTR configurations : the GT-MHR and the PB-MR reactor type designs.

As requested it is focused here on Active Magnetic Bearing (AMB) and auxiliary mechanical Catcher Bearing (CB) under normal and emergency operating conditions.

Mechanical loads as well as thermal effects are described for the GT-MHR reactor type design.

In the case of PBMR design, the work is restricted to the data available for the High Pressure Turbo-unit bearings.

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1. INTRODUCTION

This document contributes to the 5th PCRD HTR-E WP3. It aims to provide the functional requirements for axial and radial bearings designed to support the Turbo Machine (TM) in the GT-MHR reactor type design and the High Pressure Turbo-unit (HPT) in the PB-MR reactor type design.

Active Magnetic Bearing (AMB) technology is foreseen to prevent the risk of leakage and fluid ingress in the primary helium loop induced by mechanical and lubricated bearings.

Such bearing provides unique properties and high operational reliability. Basically an AMB is coupled with a mechanical auxiliary Catcher Bearing (CB).

Bearing functional requirements are requested in normal and emergency operating conditions. Parameters such as type of reactor, geometry of the TM, bearing n° and place, load and masse distribution, speed level, size, pressure, temperature, lifespan are taken into account in the frame of this document.

2. CASE OF A GT-MHR REACTOR TYPE

2.1 Turbomachine design

In the GT-MHR design the TM is an in-built, vertically arranged aggregate. All TM components (high and low pressure compressors, turbine and electric generator) are assembled on a single and long rotating shaft. Generator rotor and Turbo-Compressor (TC) rotor are connected rigidly to each other by coupling.

Six radial and one axial AMB are necessary to support the TM and to allow its rotor dynamic control. For starting and shutdown periods or at AMBs failure auxiliary CB are used complementary to the AMB.

The TM rotor design with basic dimensions and masses located in inter-support areas is depicted schematically in figure 1.

2.2 Bearing description

2.2.1 Function

AMB should be able to control the stiffness and damping of the bearings according to the rotor displacements measured by sensors. Each AMB is arranged with a CB in a single unit.

AMBs should be designed to control and maintain Turbo-Machine (TM) rotor stability and to sense radial and axial loads in all operating mode including critical rotation frequencies. Sustaining of TM is done by regulating current in AMBs with a control system.

AMBs should be able to limit the amplitude of rotor vibrations due to resonance, by its inner electromagnetic damping capability. The damping energy is brought by the power supply or by the drive.

CBs should be designed to support loads from the turbo-machine rotor when AMBs fails and during mounting, dismounting and transportation of the generator and the turbo-compressor. Moreover CBs must ensure extended coastdown under normal loads and sustain increase in short duration loads (resonance zones and external impacts).

2.2.2 Radial bearing arrangement

In the present design dimensions and parameters of all radial bearings are unified except for the top generator excitor radial bearing n°1 for which dimension and design are yet to be defined.

The axial bearing and the radial bearing n° 4 located just below it compose an axial / radial unit.

The GT-MHR reactor type TM radial bearing arrangement is depicted in figure 2 and 3.

2.2.2.1 Radial GT-MHR AMB specification

A radial AMB consists of :

- a stator,
- a rotor,
- a set of rotor radial displacement sensors,
- an electronic control system.

The present GT-MHR radial AMB stator comprise four independent electromagnets (one per quadrant) : 2 for positioning of rotor in X and 2 for positioning of rotor in Y axis. Each electromagnet could be composed of 2 or more poles.

To enhance reliability and to simplify control system coil of each coil group could be connected to parallel branches connected themselves to separate power amplifier stages.

The radial AMB rotor is a cylindrical magnetic guide composed of electromagnetic steel sheets fixed on a bush. This bush is press fitted on the TM shaft and could be fixed additionally by radial pins to preclude misaligment in the process of operation.

Each radial bearing compresses a set of radial displacement sensors[1] designed to detect the shaft position and generate a signal for the AMB control system.

Sensing elements are inductance coil mounted on a ferromagnetic core. Base members of sensors are rigidly fixed to an intermediate bush which is itself fixed to the AMB stator. Stiffness of fixing is required to ensure sensor natural frequencies to be considerably higher than the rotor operating frequencies. In addition, sensor signal could be used for diagnosis algorithms and operation monitoring.

Shaft speed sensors are installed in the machine to detect the rotor rotational speed.

Radial AMB basic dimensions are the following :

– Maximal stator core outer diameter ~1100 mm

– Minimal shaft diameter ~700 mm

– Maximal axial length of stator ~830 mm

– Radial gap between stator and rotor ≥ 2 mm

2.2.2.2 Radial GT-MHR CB description

In the present design, a radial CB is composed of 3 planetary rollers located around the rotor with a smaller clearance that the clearance with the AMB : radial gap between CB and rotor is 0,5 mm.

Each planetary rollers comprises :

- a housing,
- 3 satellites,
- an inner vertical roller located between satellites, and lined up with the housing axis.

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[1] If redundancy is required number of sensors may be doubled.

Planetary rollers ensure to reduce ball bearings rotational speed in order to improve operating conditions of CB. For instance when the radial CB is actuated at the nominal rotor rotational speed of 3000 rpm ball bearings rotational speed amounts to 1590 rpm.

The housing is a rigid Ð-shaped bush which has special belts over its outer diameter to provide contact with the rotating rotor.

A satellite comprises two ball bearings located in a bearing race and rotating around a finger rigidly fixed to the planetary roller body. The 3 satellites roll over the surface of an inner vertical shaft when the planetary roller body rotates.

Each vertical planetary roller is fixed to the radial CB housing and is secured against turning-through.

Special bearings with dry film lubrication are supposed to be used as ball bearings. Bearings should have specific coatings for the balls and race ways.

However many other technological solutions are possible.

2.2.3 Axial bearing arrangement

The axial bearing is located on the Turbo-Compressor (TC) rotor top part under the coupling interconnecting the generator rotor and the TC (see figure 1).

The GT-MHR reactor type TM axial bearing is depicted in figure 4.

2.2.3.1 Axial GT-MHR AMB specification

The axial AMB consists also of :

- a stator part,
- a rotor part,
- rotor axial displacement sensors,
- an electronic control system.

The stator part of each single effect bearing is made of annular electromagnets. Windings are inserted in grooves between the poles of each electromagnet.

As in a radial bearing each coil can be subdivided into several independent branches commuted by separated transistors controlled by common input signal[2].

The rotor part of the bearing rotor is installed between the stator electromagnets. It is a massive steel disk fastened on the TM shaft. An anti-rotation system must be designed.

Axial AMB basic dimensions are the following :

– Stator core outer diameter ~1600 mm

– Stator core inner diameter ~970 mm

– Minimal shaft diameter under the thrust disk ~470 mm

– Air gap between stator and rotor ≥ 2,5 mm

2.2.3.2 Axial GT-MHR CB description

In the present design, the axial CB is a thrust ball bearing with gas supply from a supporting system. It is located below the axial AMB. The gap between CB and rotor is 0,5 mm.

The body encapsulating the thrust bearing is supported by springs designed to reduce dynamic loads in case of a rotor drop on the CB. Load is sensed by the bearing working balls, between which separating balls are located playing the role of a lubricant.

In the present option ball should be made of silicon nitride or particular grade of steel. Separating balls surface should be covered with a lubricating material such as MoS_2.

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[2] Number of these branches is to be defined in subsequent design phases.

The axial CB has an additional gaseous system allowing to unload the pivot and to prevent a direct contact between the pivot and balls at nominal rotational speed thus removing loads on CB components and providing normal coastdown of the rotor.

In the end of coastdown, when speed gets down gas supply is interrupted and balls return to operation.

However many other technological solutions are possible.

2.2.4 Interactions between bearings and interfacing components

Interactions between bearings and interfacing components are to be handled by the following interfacing systems :

- AMBs control system,
- AMBs cooling system,
- CBs cooling system.

2.2.4.1 AMBs control system

AMBs control system shall provide assurance of maximum accuracy of shaft positioning in central position and of quick response of AMBs to external impacts.

The system provides power supply and control for the TM AMBs in all operation modes including transient and emergency modes.

2.2.4.2 AMBs cooling system

The AMBs cooling system provides removal of heat released in AMBs. Heat removal in the generator radial bearings and in the radial-axial AMB is provided by the generator cooling loop. TM lower radial AMBs are cooled by the in-circuit He circulation.

2.2.4.3 CBs cooling system

Radial CBs are cooled due to He circulation during TM coastdown. A additional cooling system could be required depending on the CB technology used.

2.3 Operating conditions

2.3.1 Operating modes

In TM operating conditions the following modes have to be taken into account :

- nominal mode,
- mode with the TM rotor rotational speed rising up to 3600 rpm,
- start up mode,
- scheduled shutdown mode,
- emergency shutdown mode,
- emergency shutdown mode at AMBs failure,
- TM jamming mode,
- operating mode under external impact conditions (Design Basis Earthquake and Maximum Design Basis Earthquake).

Issues associated with maximum displacement and a "fall" on the CBs are also to be considered.

2.3.2 TM normal operating conditions

Bearings normal operating conditions are :

- Environment : He
- Radial bearings load (10^4 N) : 10
- Axial bearing load (10^4 N) : 105
- Normal rotational speed (rpm) : 3000
- Maximum continuous speed (rpm) : 3060
- Minimum allowable speed (rpm) : 2940
- Trip speed (rpm) : 3600
- Medium pressure (MPa) : 2,64 - 4,4
- Medium temperature (°C) : 110 °C
- Lifespan (hr) : 150 000

2.3.3 TM emergency operating conditions

The He medium temperature in emergency mode could reach 450°C in a short time. However due to the fact that no specific data for duration of emergency mode are available bearings should be designed for long-term operation at this temperature.

Reduction in He pressure from 4,4 to 0,1 MPa could be done in 50-100 s.

2.4 Thermal loading

2.4.1 Thermal characteristics

Heat release in AMB windings was determined by electrical calculations. Thermal resistances were calculated from specified geometrical dimensions of structures, thermal physic characterisitics of materials and cooling He medium.

Thermal energy in the AMB is liberated both on ferromagnetic parts of stator and rotor magnetic guides as losses in steel and in electromagnets winding as losses in copper.

Winding wire should withstand service temperature of 350°C in long-term operation and up to 600°C in short term operation.

2.4.2 Radial AMBs

Thermal calculations on radial AMBs are to be done considering the two conditions below :

– P = 2,64 MPa, T = 50 °C ;

– P = 4,34 MPa, T = 110 °C.

Temperature distribution and the required cooling He flow circulation to maintain a permissible temperature state of radial AMBs in nominal mode will be determined.

2.4.3 Axial AMBs

Thermal calculations on axial AMBs are to be done considering the two conditions below :

– P = 2,64 MPa, T = 50 °C ;

– P = 4,34 MPa, T = 110 °C.

Temperature distribution and the required cooling He flow circulation with account of disc friction will be determined.

2.5 Seismic loading

Bearings should be designed with account of external effects on the power unit (earthquake, shock wave, etc.) and maintain the turbo-machine operability.

Bearings shall retain their functional capability :

- at the seismic impacts of magnitude 6 on the MSK-64 earthquake scale,
- and up to the seismic impacts of magnitude 7.

Taking into account the amplification by the buildings, that leads in a first assessment to the following acceleration on the TM in case of DBE :

– $\gamma_V = 0{,}25$ g

– $\gamma_H = 0{,}5$ g

2.6 Mechanical loading

2.6.1 Load analysis

2.6.1.1 Radial loads

Basic types of radial loads are :

– radial forces occurring due to the TM rotor unbalance,

– radial electromagnetic forces caused by eccentricity of the rotor in the generator stator (acting only on radial bearings n° 2 and n°3).

– aerodynamic radial forces (coming from the turbine and the 2 compressors - generally negligible).

2.6.1.2 Axial loads

Basic types of axial loads are :

– aerodynamic axial forces (coming from the turbine and the 2 compressors),

– load from TM rotor mass.

2.6.1.3 Resonance frequencies

The control system should assure the TM passing through natural frequencies at TM starting and stopping.

TM natural frequencies are expected to be in the range 0-50 Hz. More precisely, mechanical analysis gave the results displayed in table 1.

2.6.2 Load requirements

Loads on the TM bearings were determined in the different operating modes. They are given in table 2.

AMBs and CBs required load capacities are given in table 3.

2.6.2.1 In nominal mode and at higher rotational speeds

At nominal mode operation and operation at higher rotational speeds maximal design loads could amount to ~5.10^4 N on the generator AMBs. Load should not exceed ~$2,5.10^4$ on the other AMBs.

The main portion of these loads is due to TM unbalance. With a maximal permissible eccentricity of 10 µm at 3000 rpm load occurring due to radial unbalancing forces amount to 0,1 G.

For radial bearings nominal load capacity for each support shall be equal to ~10.10^4 N.

Nominal load capacity of axial bearing shall be equal to ~105.10^4 N.

The rotor disturbance frequencies in these modes could amount to 50 and 60 Hz respectively.

2.6.2.2 During startup and normal shutdown

During start up and normal shutdown modes design loads increase sharply due to presence of resonance zones and inefficient rates to pass them through.

For CBs maximum design load amount to ~12.10^4 N [3] on the generator radial bearing n° 3.

Nominal load capacity of axial bearing is equal to ~105.10^4 N.

The rotor disturbance frequencies in these modes are approximately in the range 20-30 Hz. AMBs properties should be used to damp the rotor oscillation and to decrease forces applied to these supports.

2.6.2.3 In emergency shutdown mode

In this mode loads are entirely supported by the CBs, including higher loads caused by the presence of resonance zones.

CBs must then ensure extended coastdown under normal loads and sustain increases in short-duration loads (resonance zones and external impact).

For radial CBs with rollers load capacity in emergency shutdown mode amount to ~12.10^4 N.

For the axial CB load could rise up to 450.10^4 N when AMBs fails.

Duration to slow down from maximum speed to 1/3 of maximum speed should not exceed 50 s.

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[3] With an evaluation dynamic coefficient equals to 10.

2.6.2.4 In TM jamming mode

Under conditions of jamming maximum load amounts to ~12,5 .10^4 N on the radial bearing n° 4.

2.6.2.5 Under conditions of external impact

Accidental loads due to earthquakes are entirely supported by the CBs.

Under conditions of external impact loads on the generator bearings n°2 and n°3 amount to ~20.10^4 N in case of DBE and ~37.10^4 N in case of MDBE.

On the other radial bearings loads could be from ~2.10^4 N up to ~10.10^4 in case of DBE and from ~3.10^4 N up to ~18.10^4 in case of MDBE.

The axial load sensed by the axial bearing at the moment of an earthquake impact could be from ~19.10^4 N ( ) to ~131.10^4 ( ) in case of DBE and from ~7,5.10^4 N ( ) to ~158.10^4 ( ) in case of MDBE.

Disturbance frequency under external impacts should be in the range of 0 to 20 Hz.

2.7 Other requirements

- Rotor part of the bearing would have to be manufactured with care, control accuracy depending closely on its circularity.
- In emergency modes with primary circuit depressurisation, during repair or start up or adjustment work, bearings should work in He, air or He-air mixture with pressure of 0,1 MPa.
- Bearings should withstand action of the turbo-machine rotor rotating at nominal speed of 3000 rpm and maintain turbo-machine operability in case of runaway rotational speed up to 20% of the nominal speed (3600 rpm).
- Bearings should have minimal mass/dimension characteristics.
- AMB and CB should be preferably designed as a single unit and have provisions for aggregate wise replacement.
- Materials used for bearings should not liberate contaminating substances in the primary circuit.
- AMB life span should not be less than 150 000 hours[4] with failure interval not less than 60 000 hours. CB must withstand up to 20 rotor "drops".
- AMB control system must ensure power supply and control in all normal operation modes

[4] With account of scheduled repair in 6-8 years

3. CASE OF A PBMR TYPE REACTOR

3.1 Power Conversion Unit design

In the PBMR design, the Power Conversion Unit (PCU) comprises two Turbo-units, a power turbine generator set, a recuperator, a pre-cooler and an intercooler. The PCU is installed in a pressure vessel sub-divided into 5 sub-systems : the manifold vessel, the generator vessel, the recuperator vessel, the pre-cooler vessel and the intercooler vessel.

The high-pressure and the low-pressure compressor units are of similar design and construction. They provide pressure in the helium cycle, and each is driven by its own turbine. Combined they make up the High-Pressure Turbo-unit (HPT) and Low-Pressure Turbo-unit (LPT).

The HPT and the LPT are situated within the PCU manifold vessel as shown in figure 5. The power turbine generator as well as the helium transport pipes and the bypass valves necessary for control of the cycle are also located in the PCU manifold vessel.

3.2 High pressure turbo-unit design

The HPT within its plug-in container is depicted in figure 6. The HPT rotor is a single shaft construction, integrating the 9-stage compressor and the double entry turbine discs.

The HPT rotor shaft is suspended by one axial and two radial electromagnetic bearings.

The HPT is inserted into fixed vertical barrels inside the manifold, accessed through the closures to allow for hermetically controlled replacement.

The compressor housing is a split-half construction and is bolted onto the inlet casing. The compressor split casing holds the electromagnetic bearings stationary shells, as well as the variable compressor stator blades with their labyrinth half sector sealing bands, to minimize rotor gap leakage.

The turbine housing is a 3-part casting mounted on the compressor counterpart, via an isolation ring which minimize heat transfer and provide high accuracy alignment of turbine rotor. Removal of the outer turbine casing exposes the turbine double disc, which can then be bolted loose and removed.

The turbine discs overhang the top bearing, for easy removal whilst improving shaft resonant frequency to be well above the running frequency. Because of the electromagnetic bearing capability to tolerate unbalance, fitting of a new turbine disc assembly may not necessarily require re-balancing of the complete shaft unit.

3.3 Bearing description

The top radial bearing is installed between the turbine and the compressor.

The lower radial bearing is installed below the compressor, just above the axial bearing.

The axial bearing is located at the bottom of the HPT at the compressor inlet channel level.

Each bearing is composed of an AMB, with an auxiliary non-lubricated roller CB as a backup. In case of AMB failure, the CB takes the load for the period of the HPT rotor rundown.

The HPT is subject to routine maintenance removal every six years or multiples of six. Bearings will have to provide functional reliability at least over this period before being overhauled.

3.4 Operating conditions

3.4.1 HPT normal operating conditions

HPT bearings normal operating conditions are :

- Environment : He
- Radial bearings load (10^4 N) : not available
- Axial bearing load (10^4 N) : 0,7
- Normal rotational speed (rpm) : 15 200
- Medium pressure (MPa) : 7
- Medium temperature (°C) : max. 150 °C
- Lifespan (hr) : min. 53 000

3.4.2 HPT emergency operating conditions

Data no available.

3.5 Thermal loading

Data no available.

3.6 Mechanical loading

As in the GT-MHR type design, the AMBs and CBs will be designed in such a way that they are capable of taking all basic loads affecting the rotor.

Basic types of load are as follows :

- radial forces, caused by the HPT rotor unbalance,
- aerodynamic axial and radial forces (coming from the turbine and the compressor - generally negligible)
- rotor weight load,
- axial and radial forces affecting the rotor under earthquake conditions.

The HPT shaft weighs approximately 700 kg that is considerably lower than the GT-MHR TM rotor. The HPT rotor runs at 15 200 rpm.

The first bending critical frequency of the rotor is expected to be 15% higher than the first overspeed running order.

3.7 Seismic loading

Bearings shall retain their functional capability :

- at 0,2 g peak ground acceleration in horizontal direction for Operating Basis Earthquake (OBE),
- and up to 0,4 g peak ground acceleration in horizontal direction for Safe Shutdown Earthquake (SSE).

4. APPLICABLE CODES AND STANDARD

Mainly RCC-M or ASME are to be used. Other references are possible but regulations and guidelines to be used in the designs are to be internationally recognised.

Components shall be designed and manufactured to an acceptable international pressure vessel code, provided this code can demonstrate its capability of meeting all the functional, safety and reliability requirements.

5. CONCLUSION

In this document functional requirements are given for bearings supporting the TM shaft in the GT-MHR design.

In the case of the PBMR reactor type design, the work is restricted to the data available for the HPT bearings. However few data are available in this case because the present design is only conceptual.

In a first assessment it should be possible to apply to the PBMR HPT the same requirements than for the GT-MHR TM in terms of fonction, reliability, control system efficiency, etc.

Beyond here-given informations on PBMR reactor type HPT bearings, data as bearings arrangement and size, thermal and mechanical loadings in normal and emergency operating conditions, resonance frequencies are not described in the present document.

Figure 1 : GT-MHR design - Radial and axial bearings location on the TM shaft

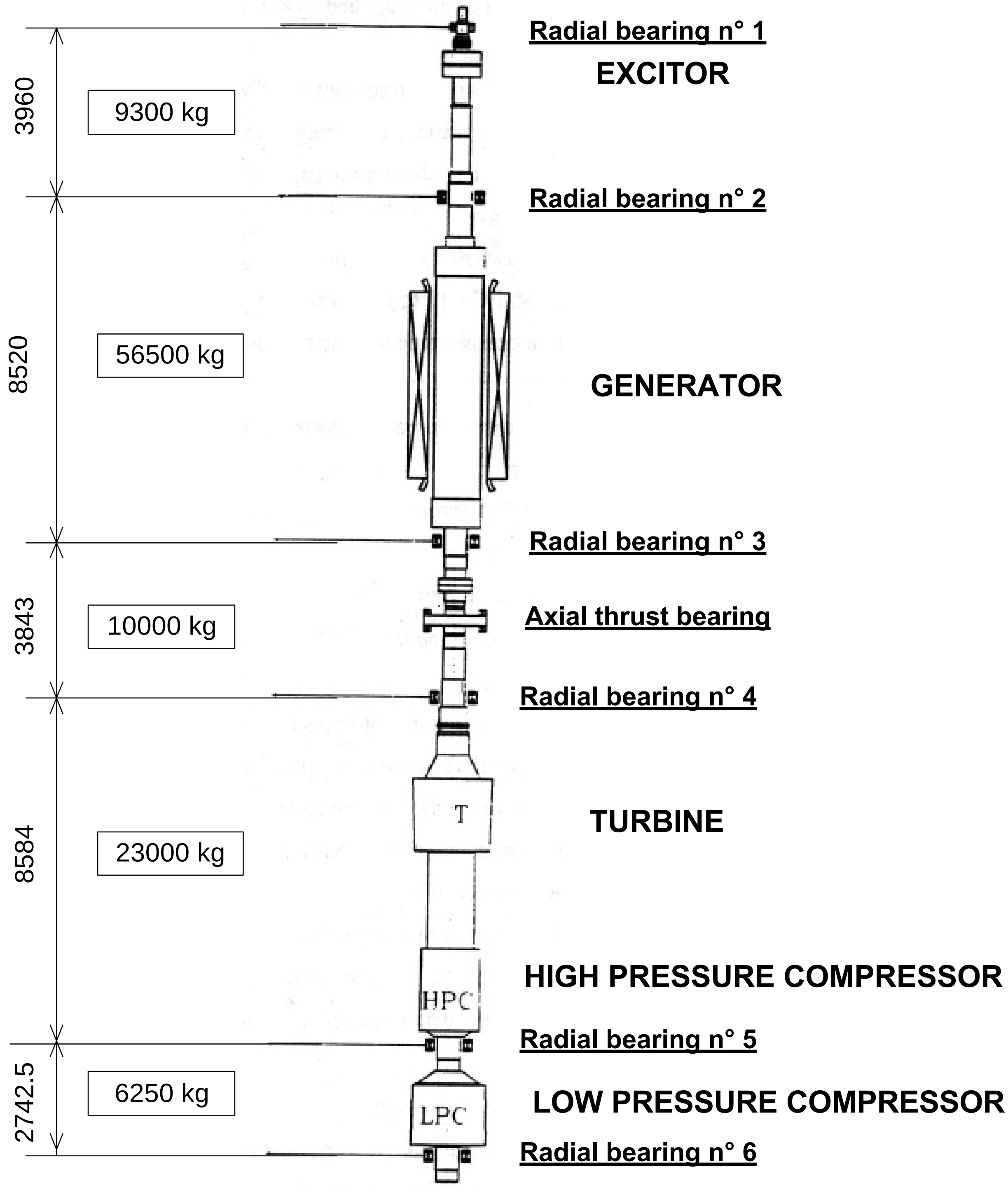

Figure 2 : GT-MHR design - Radial bearing arrangement

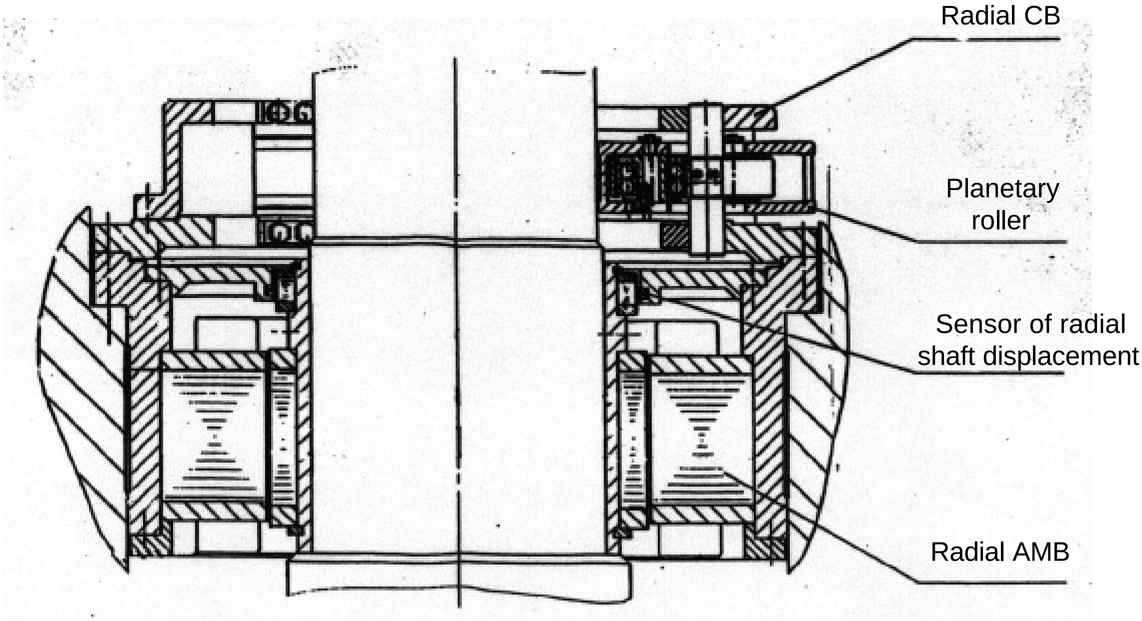

Figure 3 : GT-MHR design - Radial CB arrangement

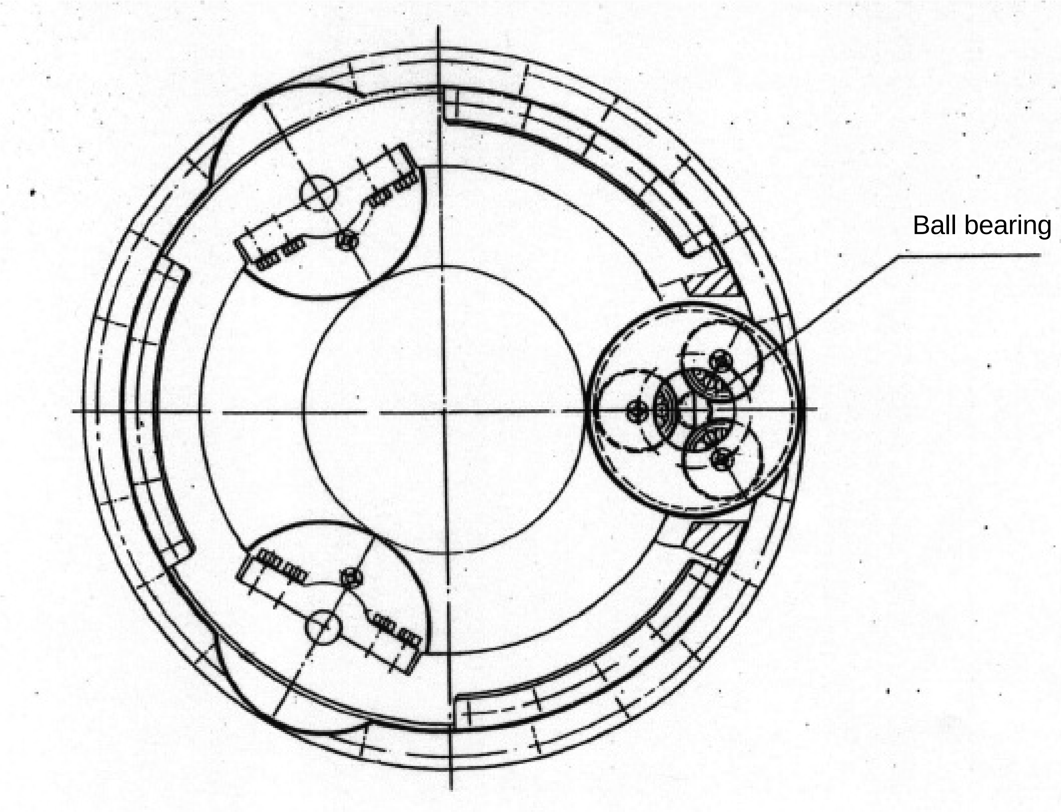

Figure 4 : GT-MHR design - Axial bearing arrangement

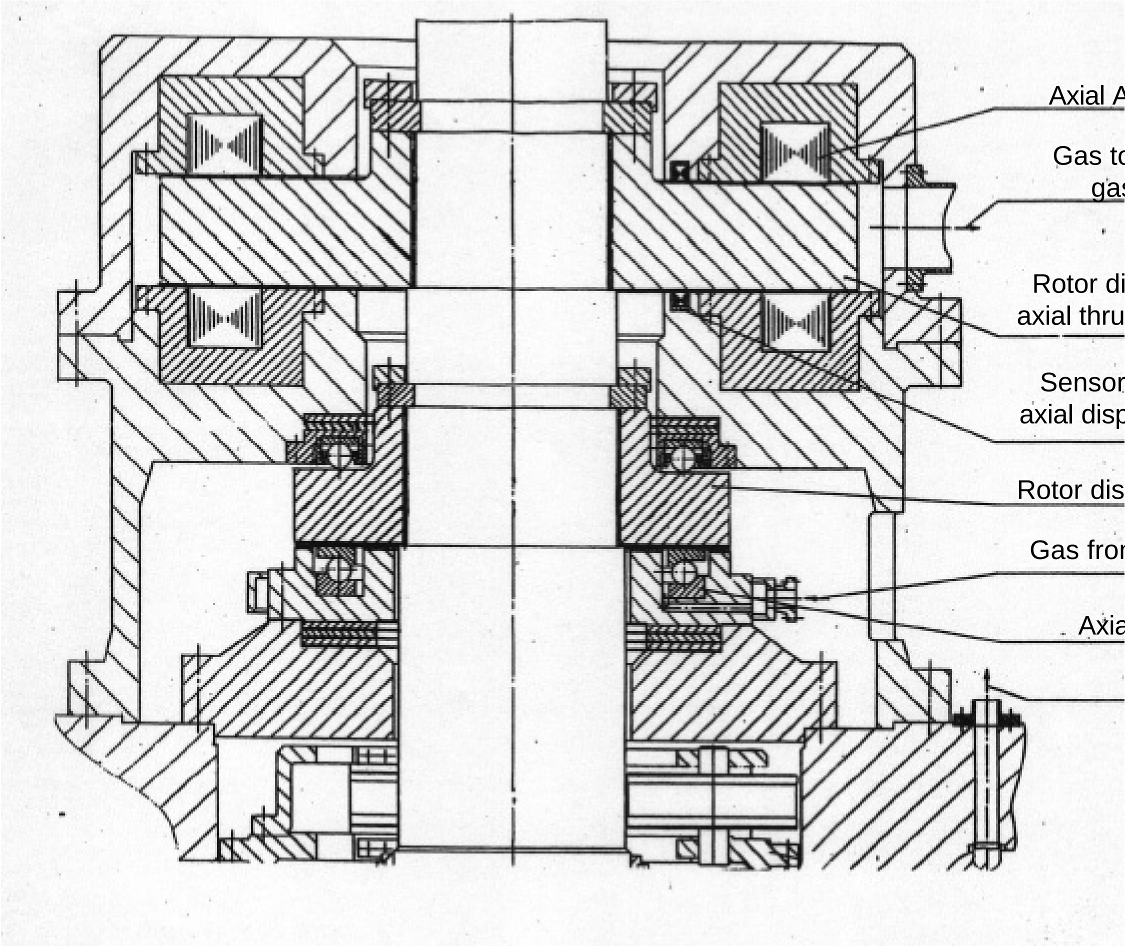

Table 1 : GT-MHR design - TM shaft natural frequencies

Configuration	Natural frequencies (Hz)
Shaft without support (stiffness 0 N/mm)	$f_1 = 3$ $f_2 = 11$ $f_3 = 16$ $f_4 = 32$ $f_5 = 47$ $f_6 = 57$ $f_7 = 85$ $f_8 = 106$
Shaft on absolutely rigid supports (stiffness →)	$f_1 = 24$ $f_2 = 26$ $f_3 = 45$ $f_4 = 71$ $f_5 = 92$ $f_6 = 110$
Shaft on support (stiffness 110 000 N/mm)	$f_1 = 9,5$ $f_2 = 11,5$ $f_3 = 16,1$ $f_4 = 23,4$ $f_5 = 28$ $f_6 = 38,3$ $f_7 = 52,5$ $f_8 = 65,4$ $f_9 = 88$ $f_{10} = 109$

Table 2 : GT-MHR design - Loading assessment onto bearing supports at various operating mode

Operating mode	Load numerical value (x 10^4 N)						
	Radial bearings						Axial bearing
	1	2	3	4	5	6	
Nominal, 3000 rpm	0,465	3,72	3,75	1,65	1,47	0,3	45 ↓
Rotational speed up to 3600 rpm	0,67	5,16	5,21	2,38	2,12	0,43	25 ↓
Start up (speeding up)	1,7	11,8	12,0	5,9	5,3	1,08	105 ↓
Scheduled shutdown	1,7	11,8	12,0	5,9	5,3	1,08	105 ↓
Emergency shutdown, when AMBs fail	0,465 [1] 1,7 [2]	3,72 [1] 11,8 [2]	3,75 [1] 12,0 [2]	1,65 [1] 5,9 [2]	1,47 [1] 5,3 [2]	0,3 [1] 1,08 [2]	45 ↓ ... 450 ↓ [1] 105 ↓ [2]
TM seizing (blocking)	0,465	3,72	3,75	12,15	5,0	0,3	45 ↓
External impact (DBE)	1,8	20,2	20,4	9,9	8,82	1,8	19 ↓ ... 131 ↓

Notes :

[1] At the moment of loss of power

[2] During TM rotor coastdown

↓ Denote the downward direction of force

Table 3 : GT-MHR design - AMBs and CBs required load capacity

Bearing type	Required load capacity (x 10^4 N)						
	Radial bearings						Axial bearing
	1	2	3	4	5	6	
AMB	2	12	12	6	6	2	105
CB	2	20	20	10	10	2	450

Figure 5 : PBMR design - Turbo-units located in the PCU manifold

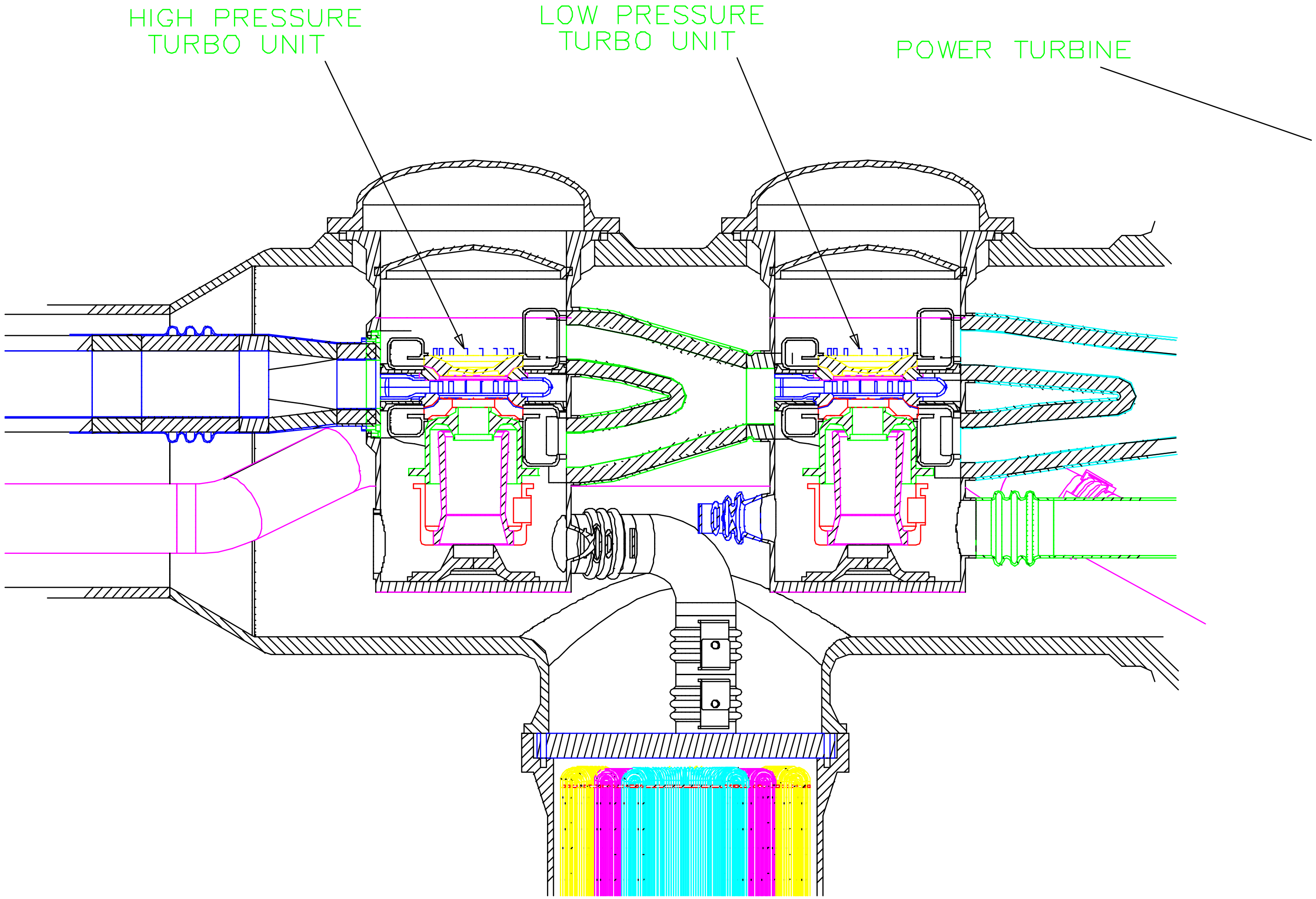

Figure 6 : PBMR design - High pressure turbo-unit in the plug-in container